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CONTENTS

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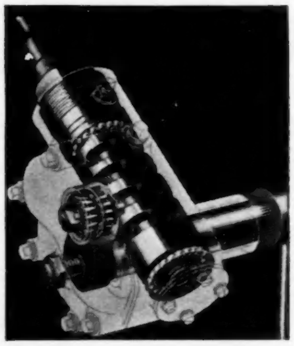

Eliminating

Crankshaft Torsional Vibration in Radial Aircraft Engines

By E. S. Taylor
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CRANKSHAFT torsional vibration has become a serious problem in aircraft engines. Thanks to much experimental work, we now have a good working knowledge of the two phases of the problem, the elastic and inertia characteristics of the crankshaft-propeller combination and the forces to which this system is subjected.

Methods used in the past to reduce vibration have been to change the elastic characteristics of the crankshaft, or to incorporate direct damping or some form of vibration damper of which the Lanchester and the resonant damper are examples. All of these methods have serious limitations. An interesting device which is capable of eliminating vibration in constant speed machinery is the undamped absorber. For variable speed machinery this absorber is of no value. By arranging an undamped absorber so that the restoring force varies with speed, it is possible, theoretically, to eliminate vibration in certain variable speed machinery. A pendulous weight rotating with the crankshaft of an engine can be constructed so that the restoring force has the desired variation with speed. The device is simple and has a number of outstanding advantages, chief of which is the extraordinary effectiveness of a comparatively small device. The principle has been utilized in a device applied to the Wright "Cyclone" series radial aircraft engines.

Torsiograph records taken on this engine show complete freedom from vibration with the device functioning.

THE constant demand for more output from aircraft engines has resulted in increased mean effective pressures and piston speeds. For two reasons, this has led to increasing trouble with crankshaft torsional vibration, especially in radial aircraft engines. In the first place, the natural damping of the crankshaft-propeller system is a function of engine size and is independent of output. Thus an increase of mean effective pressure will result in a proportionate increase in amplitude of vibration at all speeds. Secondly, with direct-drive single-row engines, increase in piston speed has resulted in operation nearer the resonant speed of the crankshaft-propeller system. The natural result of greater output has been an increasing number of scuffed propeller hub cones, battered controllable propeller-blade bearings, worn reduction gears and, in some cases, broken crankshafts and propeller shafts. The seriousness of the situation has led to a large amount of experimental and analytical work, thanks to which we now have a good background knowledge of the problem.

[This paper was presented at the Annual Meeting of the Society, Detroit, Jan. 17, 1936.]

For a complete understanding of the phenomenon, we must have a knowledge of its two phases—first, the elastic and inertia characteristics of the crankshaft-propeller combination and, second, a knowledge of the nature of the forces to which the elastic system is subjected.

In order to represent mathematically the characteristics of the elastic system comprising the crankshaft and propeller of a radial aircraft engine, it is necessary to indulge in considerable simplification. It has been shown by many experimenters that the vibration characteristics of the crankshaft-propeller system of a radial engine can be reproduced approximately by the system shown diagrammatically in Fig. 1. I is a flywheel whose moment of inertia is equal to the moment of inertia of the crankshaft about its axis plus Mr^2, where M is the mass necessary at the crankpin to balance the crankshaft and counterweights statically, and r is the crank radius. S is a shaft whose torsional stiffness is

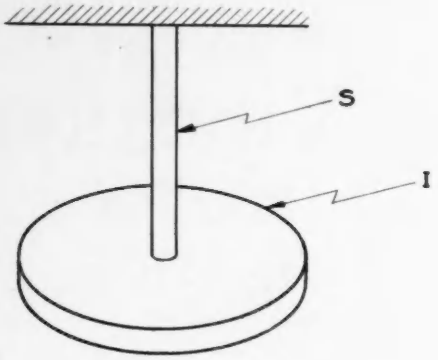

Fig. 1—Torsional Vibrating System Equivalent to Crankshaft

equal to the torsional stiffness of the crankshaft from the rear of the propeller hub to the center of the crankpin.

It will be noted that the shaft S is considered as fixed at its upper end. This is equivalent to considering the propeller as a rigid body of infinite moment of inertia. The use of controllable pitch propellers which have relatively flexible connections where the blades join the hub, has made this approximation less accurate and has somewhat confused the problem, since different propellers may be used with the same engine, resulting in different vibration characteristics.

For greater ease of representation, the torsional system of Fig. 1 may be replaced by the mathematically equivalent linear system of Fig. 2, consisting of a mass, M, supported by a spring K, which is fixed at its upper end. It will be noted that a damper, C, has been added in Fig. 2. The damper may be considered as a plunger in a cylinder filled with oil, which offers resistance proportional to the velocity of the mass, M. By assigning a proper value to the proportionality constant of the damper, the system of Fig. 2 will have very nearly the same vibration characteristics as a radial engine crankshaft and propeller combination, and under the action of the force $F(t)$, it will perform the same sort of oscillation.

If the system illustrated in Fig. 2 be subjected to a periodically varying force

$$F(t) = F_o \sin \omega t \tag{1}$$

where F_0 and ω are constants, and t represents time, after a certain time interval the system will settle down to a steady oscillation whose displacement from the equilibrium position

$$x = x_o \sin (\omega t - \phi) \tag{2}$$

where x_0 and ϕ are constants. Let us define $x_{stat.}$ as the static deflection under the force, F_0, and ω_0 as the frequency of free vibration of the system without damping. After selecting a suitable value of the damping constant, we may plot the relative amplitude of vibration against the ratio of the "forcing frequency", ω, to the natural frequency ω_0. This gives the familiar "resonance curve" shown in Fig. 3. If we subject the system to two or more "forcing functions" simultaneously, the response will be the sum of the responses of the several forcing functions taken separately, *as long as the vibrating system is linear* (i. e., the restoring force is proportional to the displacement, the damping force is proportional to the velocity, or is very small compared to the elastic and acceleration forces and the mass is constant). Fortunately, the condition of linearity is well satisfied by the crankshaft-propeller system and, therefore, if the engine torque, T, can be expressed as a sum of terms similar to the right-hand side of equation (1), then the maximum amplitude of vibration of a crankshaft can be obtained analytically by summing the amplitudes of vibration due to each of the various components of the torque. Since the torque is of the periodic nature, it may be represented as a Fourier's series

$$\frac{\text{Instantaneous Torque}}{\text{Indicated Mean Torque}} = 1 + a_{.5} \sin (\tfrac{1}{2}\theta - \alpha_1) + a_1 \sin (\theta - \alpha_2) + a_{1.5} \sin \left(\frac{3}{2}\theta - \alpha_3\right) + a_2 (\sin 2\theta - \alpha_4) + \dots \tag{3}$$

The constants $a_{.5}$, a_1, $a_{1.5}$, etc., and α_1, α_2, α_3, etc., completely describe the variation of torque with θ, the crank angle from top center. The fractional coefficients are necessary because the torque cycle of a single-cylinder four-stroke engine is completed only after two revolutions. The form of Fourier's series is convenient for two reasons: first, because the terms are harmonic (proportional to the sine of an angle) and therefore make easy the solution of the equation of the vibrating system, and second, because of the ease of addition of torque from a number of cylinders if the torque from one cylinder is known.

In radial aircraft engines the torque due to gas pressure is chiefly responsible for the torsional vibrations of the crankshaft. A study of this torque variation for a single-cylinder engine was undertaken in the M.I.T. Automotive Engine Laboratory. Fig. 4 shows the value of the coefficients $a_{.5}$, a_1, $a_{1.5}$, etc., obtained from this work for one set of operating ing conditions. From the study of the results at many different operating conditions (variation of throttle, compression ratio, spark advance, and mixture ratio) it was found that the coefficients from a_3 to a_{10} followed closely the relation

$$a_n = \left(\frac{6.3}{n^2}\right)\left(\frac{c.r}{5}\right) \tag{4}$$

where *c.r.* is the compression ratio. This relation is also plotted in Fig. 4. In adding torque from the various cylinders of a multi-cylinder engine if the explosions are evenly spaced, the terms with subscripts which are not $N/2$ cancel each

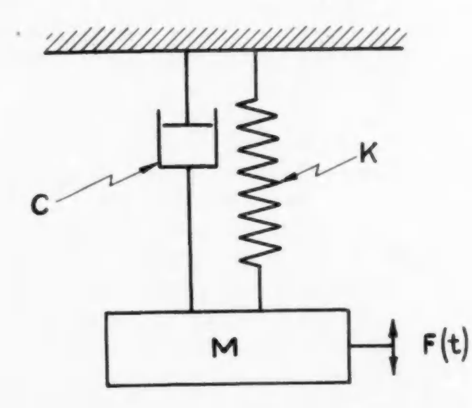

Fig. 2—Linear Vibrating System Equivalent to Crankshaft

other. (N is the number of cylinders). The remaining terms of equation (3) give the torque for the complete engine if the total engine displacement is used. The values of the phase angles, α_1, α_2, α_3, etc., vary widely with engine conditions, such as fuel-air ratio and spark advance. Since they are unnecessary in this discussion, description of their variation will be omitted. Equation (4) shows that coefficients of the higher terms of the Fourier's series representing torque are inversely proportional to the square of their harmonic order. This is interesting, since is characteristic of the coefficients of a mathematical relation with a discontinuous slope (i. e., sharp peaks). Since we know the torque varies smoothly, the relation of equation (4), if correct for the lower harmonics, will give values slightly too high for the higher harmonics. However, it fits the observed data sufficiently well.

Having an approximation of the elastic and mass characteristics of the crankshaft-propeller system, and the torque[1] in the form of a Fourier's series, we may predict, with sufficient accuracy, the behavior of the crankshaft in torsional vibration. Each of the terms of the Fourier's series representing the torque will cause a resonance curve similar to that in Fig. 3, whose amplitude depends upon the mean torque and the value of the coefficient a. The sum of all these curves (vector sum, since the maxima do not necessarily occur at the same time) will give the maximum amplitude of vibration of the shaft. When this amplitude is added to the steady deflection caused by the mean torque the result is the twist in the shaft from which the stress may be estimated.

Fortunately, in the radial engine there is only one term of the series which has a large value, and at the same time has the proper frequency to cause resonance in the operating range. This term is the one which has the same frequency as the cylinder explosions, $N/2$ per revolution. Thus in the radial engine we observe only a single resonance curve.

The mean torque of an engine driving a propeller is approximately proportional to the square of the rotative speed. Taking a nine-cylinder radial engine as an example, this means that the value of the term of the Fourier series $a_{4.5}$ $\times$ mean torque will also vary with speed and if we plot maximum amplitude, Θ_0, against r.p.m., the resonance curve will appear as shown in Fig. 5. It can be seen from Fig. 5 that the part of the twist in the crankshaft due to the 4½ order torque is tremendously multiplied by resonance of the crankshaft propeller system. Experience indicates that, for direct-drive engines, the twist will be about 15 times the twist caused by a constant torque equal to $a_{4.5}$ $\times$ mean torque. Geared engines usually have somewhat more damping, the multiplication being about 10 to 15 times. A reasonably accurate idea of the stress in the crankshaft at the resonant speed can be found very rapidly from Fig. 4.

$$\frac{\text{Stress at resonance}}{\text{Static stress}} = \frac{1_0 + 15 \times a_{4.5}}{1_0 + a_{4.5}} = \frac{1 + 5.7}{1 + 0.38} = 4.9$$

Thus the stress at resonance in a nine-cylinder direct-drive engine will be roughly five times the stress from a steady application of the maximum torque. When it is recalled that this is a *repeated* stress, and that at 1800 r.p.m., 1,000,000 stress cycles are completed in just over two hours, it is not surprising that failures have occurred. Fig. 6 gives a graphic picture of the torque in a hypothetical crankshaft at resonance. Curve a) shows the magnitude of the torque applied to the crankshaft due to gas forces at various crank angles. Curve b) shows the torque in the shaft at resonance. Note the large negative values. The negative torque is especially destructive to reduction gearing and propeller hub cones. The methods of reducing the amplitude of torsional vibration, in the past, fall into two general classifications.

(1) Changing the stiffness of the shaft so that resonance occurs outside the operating range, and

(2) Providing additional damping.

The first of these methods is somewhat limited in its application. Since the frequency of free vibration of any vibrating system varies with the square root of the stiffness, it is necessary to make relatively large changes in the stiffness of the system in order to make a change in its natural frequency and it is therefore difficult to remove the resonant speed completely from the operating range. The increase in weight required to accomplish this has led to a number of compromise designs where the resonant speed was in the lower part of the operating range and restrictions were placed upon the operation of engines at low speeds. However, operating restrictions are often neglected and are always undesirable. Furthermore, the necessity of passing through the resonant speed may ultimately result in trouble, particularly if there is frequent rapid acceleration.

Increasing the damping of the system is an effective way of reducing vibratory stresses. It is difficult to incorporate a damping device which is connected to two parts of the

[1] In certain engines, notably those with three or fewer cylinders on one crank, the torque variation due to the inertia of the reciprocating parts must also be considered.

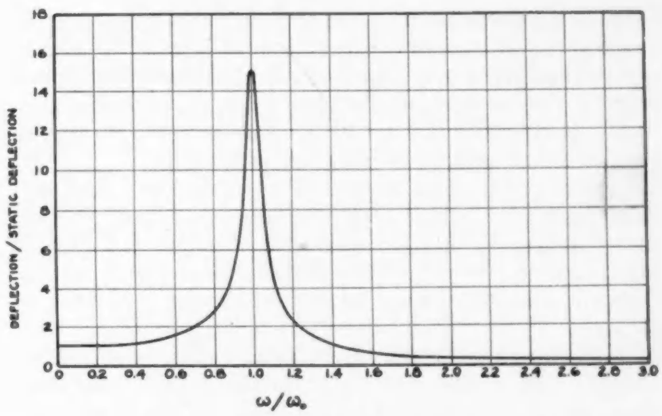

Fig. 3—Response of a Crankshaft to a Harmonic Vibrating Force

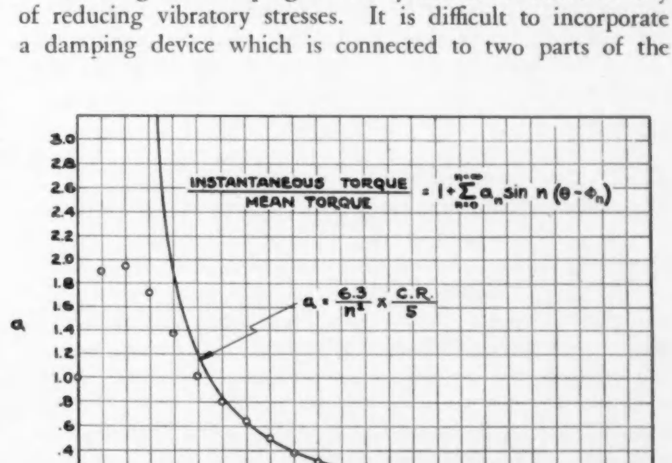

Fig. 4—Fourier Coefficients for Otto Cycle Engine Torque

crankshaft-propeller system which have relative motion, because of space limitations. Because of resulting complication and increased weight, this method has been applied only to experimental installations. Another method of providing damping is the use of a vibration absorber such as the Lanchester damper or the resonant damper. Fig. 7 shows diagrammatically the principle of the Lanchester and resonant types of vibration dampers. A particularly excellent study of these devices is included in *Mechanical Vibrations* by J. P. den Hartog, McGraw-Hill Book Co., 1934, pp. 103-117. The effectiveness of such devices is a function of the ratio of the main moment of inertia to the absorber moment of inertia, and may be expressed in terms of the ratio of the amplitude at resonance divided by the static amplitude. Fig. 8 is a plot taken from the above reference, of this relation, for the best possible arrangement of damping and spring constant. It will be noted that the resonant damper is very much more effective than the Lanchester damper. From Fig. 8 it is apparent that even with the most effective arrangement it is necessary to incorporate a large part of the inertia of the system in the damper in order to get enough reduction of the vibration to be of any practical

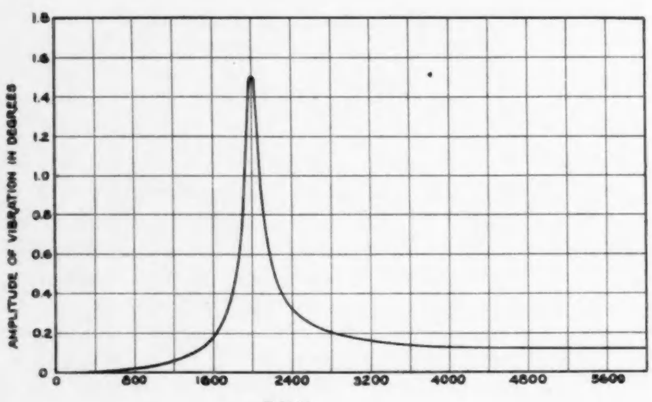

Fig. 5—Amplitude of Vibration

value. It is possible to mount the counterweights on pivots, and to provide springs and damping to constitute a resonant damper, but the complication is considerable and the effect disappointing due to the difficulty of getting a sufficient fraction of the moment of inertia in the damper. An effective damper would require a considerable addition to the engine weight.

The resonant absorber without damping, Fig. 9, is worthy of our attention. The addition of the small mass, m, elastically supported from the main mass, M, results in a complete change in the characteristics of the resonance curve. Fig. 10 shows the characteristics of the system with and without such an absorber. It will be noted that we now have two resonant frequencies, but that *there is one frequency at which there is no vibration*. This frequency is always equal to the natural frequency of the absorber, which may be defined as the frequency of free vibration of the absorber mass with the main mass held stationary. The absence of vibration is due to the fact that any motion of mass, M, at the natural frequency of the absorber will cause the mass, m, to vibrate with infinite amplitude. However, the force acting on the main mass, M, due to the spring in the absorber will be 90 deg. out of phase with the motion of the mass, and consequently

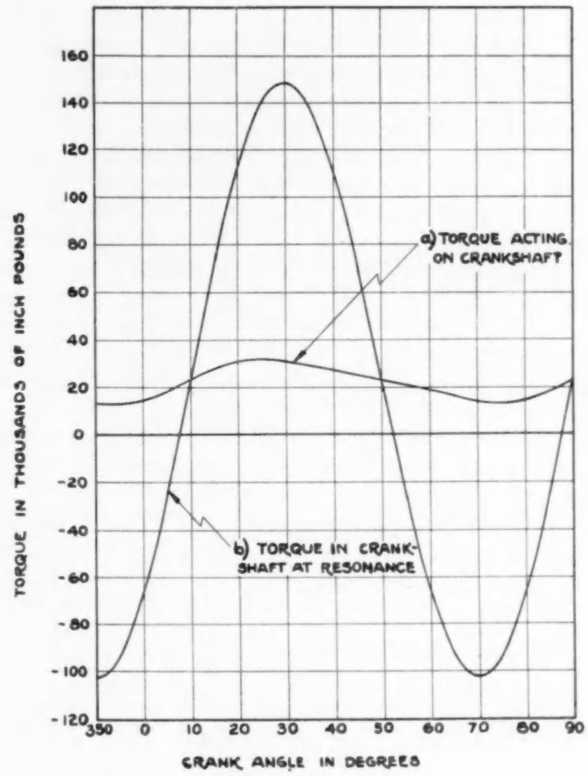

Fig. 6.

will have a maximum effect in removing energy from the main mass and reducing its motion. If the force causing vibration is finite, the amplitude of the mass, m, will increase to a point where the force in the spring, k, will exactly balance the external force acting on the mass, M, *provided the external force is harmonic and of a frequency equal to the natural frequency of the absorber.* For constant speed machinery, the device is very effective, but for variable speed devices it is worse than useless since with the absorber we have two resonant frequencies against one without. The undamped absorber suggested to the author an even simpler arrangement which has proved remarkably effective in suppressing torsional vibration in the radial engine. If the

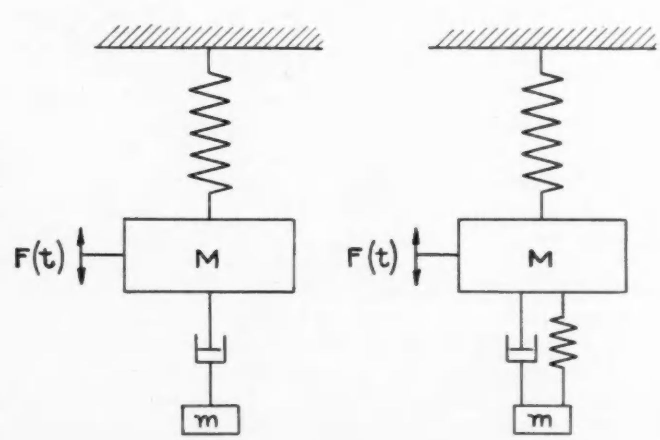

Fig. 7.

stiffness of the spring, k, could be varied so that the natural frequency of the absorber would always be equal to the frequency of the force causing vibration, we would have an absorber which would completely eliminate vibration at all speeds.

The frequency of the forcing function, in the case of an engine, is a constant times the speed of rotation of the crankshaft. We desire an absorber whose frequency will always equal the frequency of the forcing function. This will require that the restoring force of the absorber vary with the square of the r.p.m. If, instead of depending upon springs, we make the restoring force depend upon centrifugal force, which varies with the square of the r.p.m., it would appear that we are on the right track.

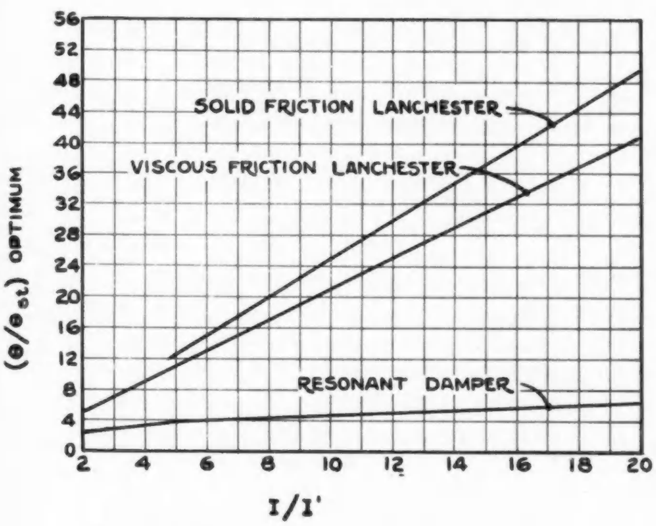

Fig. 8—Performance Curves for Various Dampers

Let us consider the vibrating system shown in Fig. 11, consisting of a pendulum, P, attached to an arm, D, rotating in a horizontal plane at a constant speed. From the geometry of Fig. 11,

$$r_1 \sin \phi = L \sin (\theta - \phi)$$

when the angle is small

$$\phi = \frac{L}{r_1}(\theta - \phi)$$

whence

$$\theta - \phi = \frac{r_1 \theta}{r_1 + L}$$

Since $r_1 + L = r_2$ (for small angles),

$$\theta - \phi = r_1 \theta / r_2 \qquad (5)$$

In addition to the notation defined in Fig. 11, let

F = centrifugal force acting on P
I = moment of inertia of P about C
m = mass of P
N = speed of rotation of D
$\Omega = 2\pi N$
n = frequency of free vibration of P
$\omega = 2\pi n$
$$F = m\Omega^2 r_2 \qquad (6)$$

The component of this force at right angles to the length of the pendulum tends to restore the pendulum to its equilibrium position and is equal to

$$F \sin (\theta - \phi) \text{ or, for small angles,}$$

$$F (\theta - \phi)$$

For a constant speed of rotation of the arm, D, the equation of motion of the pendulum is

$$I \ddot{\theta} + F (\theta - \phi) L = 0$$
$$(\dot{\theta} = d\theta/dt, \ddot{\theta} = d^2\theta/dt^2).$$

Substituting the value of $(\theta - \phi)$ from equation (5), and the value of F from equation (6),

$$I \ddot{\theta} + m\Omega^2 r_2 L \left(\frac{r_1}{r_2}\right) \theta = 0 \qquad (7)$$

This equation is satisfied by

$$\theta = \theta_o \sin \omega t$$

Substituting this in equation (7), we obtain

$$-I \omega^2 \theta_o \sin \omega t + m\Omega^2 r_2 L \left(\frac{r_1}{r_2}\right) \theta_o \sin \omega t = 0$$

Whence

$$\omega^2 = m\Omega^2 r_1 L/I$$

For a simple pendulum $I = m L^2$, and $\frac{\omega}{\Omega} = \sqrt{\frac{r_1}{L}}$

since $\omega = 2\pi n$

and $\Omega = 2\pi N$

$$\frac{n}{N} = \sqrt{\frac{m r_1 L}{I}}, \text{ or for the simple pendulum,} \qquad (8)$$

$$\frac{n}{N} = \sqrt{\frac{r_1}{L}}$$

Therefore, the small pendulum of Fig. 11 has the desirable property of a resonant speed proportional to the speed of rotation of the shaft, and thus can be designed to balance one term of the Fourier's series representing torque.

We could have arrived at the same conclusion from a di-

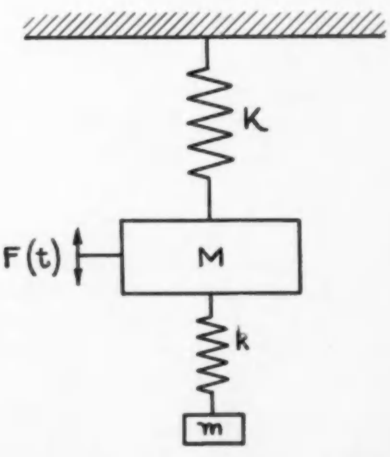

Fig. 9—Undamped Vibration Absorber

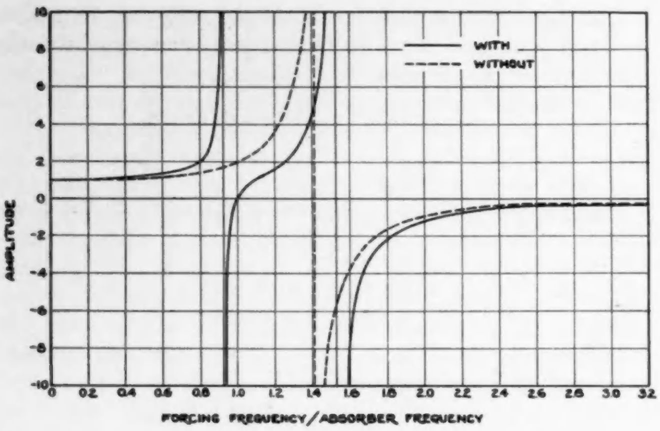

Fig. 10—Response With and Without Undamped Absorber

mensional analysis of the device in Fig. 11. Variables which might affect the frequency, n, of the pendulum are

Variable	Symbol	Dimensions
Rotative speed	N	1/Time
Mass. of pendulum	M	Mass
Length of pendulum	L	Length
Moment of Inertia	I	Mass $\times$ Length^2
Ratios describing the other dimensions of the device	R_1, R_2, R_3, etc.	0
Frequency of pendulum	n	1/Time

These variables may be arranged in dimensionless products as follows:

$$\frac{N}{n}, \frac{M L^2}{I}, \text{ and } R_1, R_2, R_3, \text{ etc.}$$

From dimensional analysis we may state that

$$\frac{N}{n} = f\left(\frac{M L^2}{I}, R_1, R_2, R_3, \text{etc.}\right)$$

This is the same form as equation (8). If the function is the square root and if

$$R_1 = \frac{r_1}{L} \text{ and } f(R_2, R_3, \text{etc.}) = 0$$

it will be identical.

As a further check upon the foregoing reasoning, let us set up the mathematical expressions for a vibrating system similar to a crankshaft-propeller combination with a pendulous vibration absorber. Fig. 12 shows such a system. In order to be perfectly general we shall, in this case, include the effect of a finite propeller.

The shaft S which may be assumed to rotate in fixed bearings (not shown) carries a pair of discs D and D' representing the inertia of the engine parts and the propeller respectively. At the point C, distant r_1, from the axis of the shaft, a pivot provides a support for the small pendulum, P.

A periodic torque, T, is applied to the disc, D. O-O is a fixed, vertical datum line or axis of coordinates, passing through the axis of shaft, S, in the plane of disc, D, and O'-O' is a similar vertical datum line in the plane of disc D'.

Characters employed in the discussion may be defined as follows:

θ_1 is the angular displacement of the pendulum P, relative to the fixed line O-O,

θ_2 is the angular displacement of the disc D, relative to the fixed line O-O,

θ_3 is the angular displacement of the disc D' relative to the fixed line O'-O',

I_2 is the moment of inertia of disc D,

I_3 is the moment of inertia of disc D',

m is the mass of the simple pendulum, P,

r_1 is the radius from the axis of shaft S to the pivot point of the pendulum,

r_2 is the radius from the axis of shaft S to the center of gravity of pendulum,

k is the stiffness coefficient of shaft S,

T is the periodic torque applied to disc D,

Ω is the mean angular speed of disc D.

If, under the conditions above assumed, the pendulum bob, P, be displaced a small distance from its normal position, the restoring torque from equation (7) may be expressed as $-m\Omega^2 r_1 L (\theta_2 - \theta_1)$. An opposite torque, $m\Omega^2 r_1 r_2 (\theta_2 - \theta_1)$ is exerted upon disc D by the pendulum and the bearing supporting the shaft.

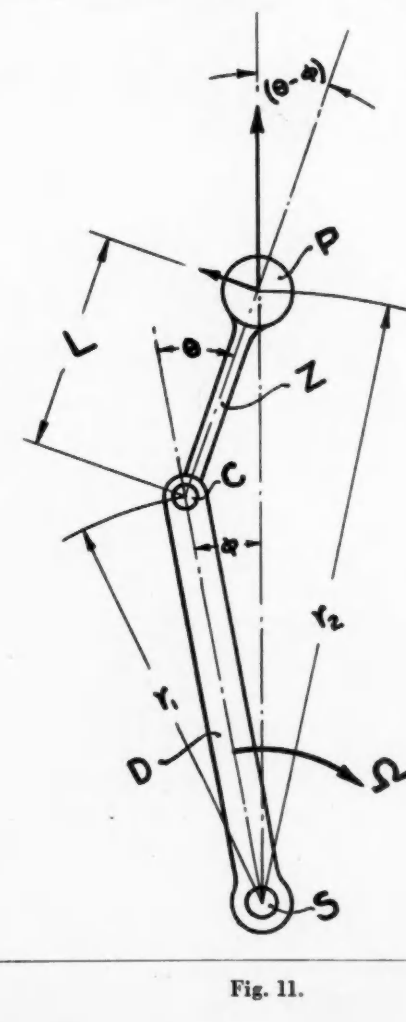

Fig. 11.

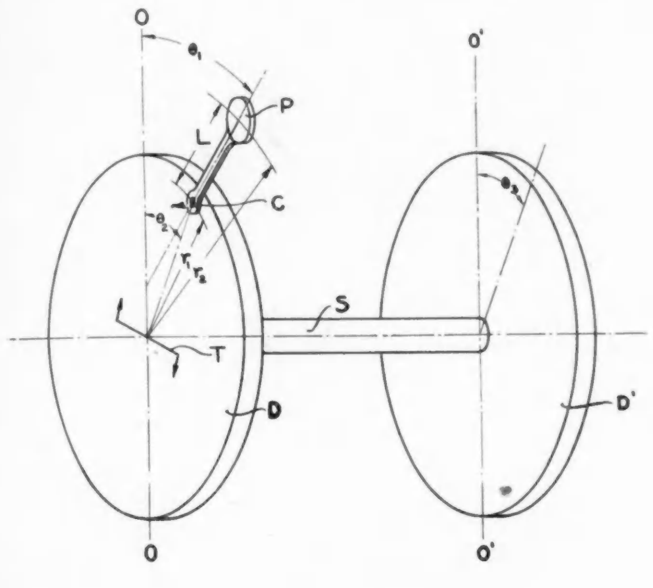

Fig. 12.

Assuming, for example, that the periodic torque, T, which acts on disc D varies in accordance with the equation $T = T_0 \sin \omega t$, where T_0 is the maximum value of the torque and t is the time in seconds, the equations of motion may be written as follows:

$$m L^2 \ddot{\theta}_1 + m\Omega^2 r_1 L (\theta_1 - \theta_2) = 0$$

$$I_2 \ddot{\theta}_2 - m\Omega^2 r_1 r_2 (\theta_1 - \theta_2) + k(\theta_2 - \theta_3) = T_o \sin \omega t$$

$$I_3 \ddot{\theta}_3 - \frac{k}{\cancel{I_3}}(\theta_2 - \theta_3) = 0$$

These may be written

$$\ddot{\theta}_1 + \frac{\Omega^2 r_1}{L}(\theta_1 - \theta_2) = 0 \tag{9}$$

$$\ddot{\theta}_2 - \frac{m\Omega^2 r_1 r_2}{I_2}(\theta_1 - \theta_2) + \frac{k}{I_2}(\theta_2 - \theta_3) = \frac{T_o \sin \omega t}{I_2} \tag{10}$$

$$\ddot{\theta}_3 - \frac{k}{I_3}(\theta_2 - \theta_3) = 0 \tag{11}$$

The above equations may be reduced to equations in two variables, as follows:

$$\text{Let} \quad Q_1 = \theta_1 - \theta_2$$
$$Q_2 = \theta_2 - \theta_3$$

whence

$$\ddot{Q}_1 = \ddot{\theta}_1 - \ddot{\theta}_2 \qquad \ddot{Q}_2 = \ddot{\theta}_2 - \ddot{\theta}_3 \tag{12}$$

Subtracting equation (10) from equation (9)

$$\ddot{\theta}_1 - \ddot{\theta}_2 + \left[\frac{\Omega^2 r_1}{L} + \frac{m\Omega^2 r_1 r_2}{I_2}\right](\theta_1 - \theta_2) - \frac{k}{I_2}(\theta_2 - \theta_3) = -\frac{T_o \sin \omega t}{I_2}$$

Substituting Q in place of $(\theta_1 - \theta_2)$, etc.

$$\ddot{Q}_1 + \left[\frac{\Omega^2 r_1}{L} + \frac{m\Omega^2 r_1 r_2}{I_2}\right] Q_1 - \frac{k}{I_2} Q_2 = -\frac{T_o \sin \omega t}{I_2} \tag{13}$$

Similarly, subtracting equation (12) from equation (11) and substituting,

$$\ddot{Q}_2 - \frac{m\Omega^2 r_1 r_2}{I_2} Q_1 + \left[\frac{k}{I_2} + \frac{k}{I_3}\right] Q_2 = \frac{T_o \sin \omega t}{I_2} \tag{14}$$

As there is no damping in the system, the forced vibrations expressed by equations (13) and (14) will be of the form,

$$Q_1 = A_1 \sin \omega t \tag{15}$$

$$Q_2 = A_2 \sin \omega t \tag{16}$$

Substituting the values from (15) and (16) in (13)

$$-A_1 \omega^2 \sin \omega t + \left[\frac{\Omega^2 r_1}{L} + \frac{m\Omega^2 r_1 r_2}{I_2}\right] A_1 \sin \omega t - \frac{k}{I_2} A_2 \sin \omega t = -\frac{T_o \sin \omega t}{I_2}$$

Dividing by $\sin \omega t$ and collecting terms,

$$\left[\frac{\Omega^2 r_1}{L} + \frac{m\Omega^2 r_1 r_2}{I_2} - \omega^2\right] A_1 - \frac{k}{I_2} A_2 = -\frac{T_o}{I_2} \tag{17}$$

Similarly, from equation (14) is obtained

$$-\frac{m\Omega^2 r_1 r_2}{I_2} A_1 + \left[\frac{k}{I_2} + \frac{k}{I_3} - \omega^2\right] A_2 = \frac{T_o}{I_2} \tag{18}$$

Solving equations (17) and (18) simultaneously, and reducing,

$$A_2 = \frac{\frac{T_o}{I_2}\left[\frac{\Omega^2 r_1}{L} - \omega^2\right]}{\left[\frac{\Omega^2 r_1}{L} + \frac{m\Omega^2 r_1 r_2}{I_2} - \omega^2\right]\left[\frac{k}{I_2} + \frac{k}{I_3} - \omega^2\right] - \left[\frac{m\Omega^2 r_1 r_2}{I_2}\right]\frac{k}{I_2}} \tag{19}$$

Now $A_2 = 0$ whenever the numerator of equation (19) becomes equal to zero, or whenever $\Omega^2 r_1/L = \omega^2$.

From equation (16) is is evident that whenever $A_2 = 0$, then (since $\sin \omega t$ does not in general equal zero), $Q_2 = 0$. However, Q_2 is the twist in shaft S. Therefore, the expression $Q_2 = 0$ represents a condition for no periodic twist in shaft S. Since the system cannot vibrate without periodic twist in the shaft, S, it follows that the vibration has been eliminated. A similar analysis is possible for a compound

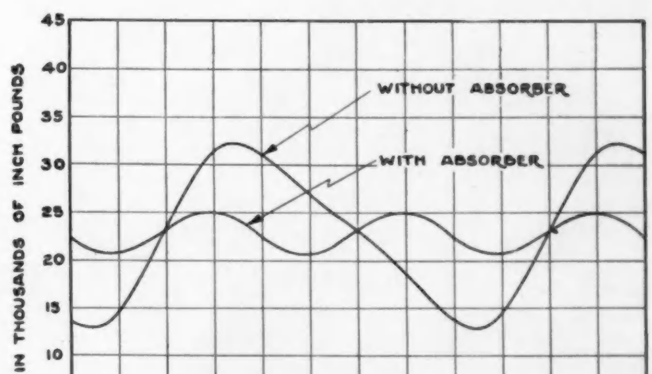

Fig. 13—Resultant Torque Acting on Crankshaft

pendulum. Pendulous arrangements in which the relation between rotation and translation is more complicated than in the compound pendulum may be analysed easily by somewhat more advanced methods.

For a nine-cylinder radial engine, if we provide a pendulous device in which the ratio $r_1/L = (4.5)^2$ we shall neutralize that portion of the torque variation equal to $a_{4.5}$ x Disp., provided the device is made sufficiently large so that its amplitude has a reasonable value. It is interesting to note that the quantities I_2, I_3 and k do not appear in the result, therefore, the operation of the device is independent of the vibration characteristics of the system to which it is attached and is dependent only upon the nature of the torque impulses. This is a very great advantage since the vibration characteristics of the crankshaft-propeller system are dependent to some extent upon the propeller, over which the engine manufacturer has no control. Also, the same device may be used without modifications for either geared or direct-drive engines.

The effectiveness of the device is such that only a portion of one counterweight is necessary to make a sufficiently large pendulum, hence it is not necessary to increase the weight of the engine. It should be remembered, however, that the balancing torque available from the pendulum is proportional to the square of the rotative speed, and if large torques are required at low speeds, it is necessary to provide a relatively large pendulum. Even in the aircraft engine, the torque at low speeds may be large during acceleration, and a pendulum larger than that required for steady operation is desirable.

Another advantage of the device lies in the fact that the torque acting on the crankshaft is reduced, due to the cancellation of the $4\frac{1}{2}$ order term of the Fourier's series (for a nine-cylinder engine) and the stress in the shaft may be less than the stress computed statically from the maximum torque. Fig. 13 shows the resultant torque acting on a crankshaft with

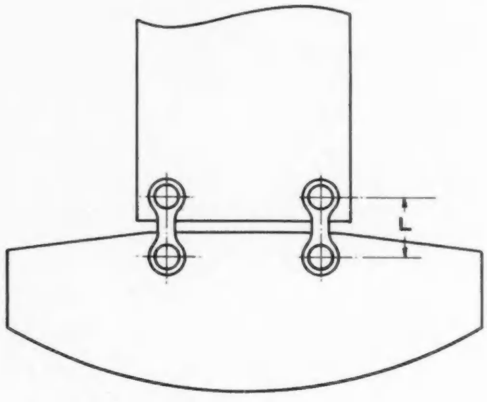

Fig. 15.

and without the absorber. An approximate calculation from Fig. 4 (neglecting higher order terms) indicates that the stress in the crankshaft due to torque will be about $1/1.38 = 72$ per cent of the stress calculated statically, or only $\frac{1}{1 + 15 \times 0.38} = 15$ per cent of the stress at resonance without the device. The torque acting on the shaft with the absorber is nearly equal to the torque from an 18-cylinder engine of equal total displacement. The torque acting on the propeller is also approximately equivalent to the torque of an 18-cylinder engine. This is sufficiently smooth so that there is little advantage to be gained by using torsionally flexible couplings in the drive for the purpose of further relieving propeller stresses. It would be simpler to employ another pendulous absorber to remove the ninth order torque if this were considered desirable. In certain cases, notably in line engines, the use of more than one device may indeed be desirable.

Since the pendulum balances the $4\frac{1}{2}$ order torque variation, vibration at this frequency is not possible with a single node or with two or more nodes in the crankshaft. Other advantages of the device over the conventional methods of damping are as follows: No energy is converted in the process of eliminating the vibration. While this energy is a negligible fraction of the engine output, it can cause considerable trouble when converted into heat inside the crankcase. There are no springs and the adjustment of the device is not affected by changes in temperature or by wear, as in the case of viscous and solid friction dampers.

The almost unbelievable number of desirable features of this type of vibration absorber led the Wright Aeronautical Corporation to undertake to incorporate one in the "Cyclone" engine. Two difficulties arose—first, it is not possible to achieve the desired frequency of $4\frac{1}{2}$ cycles per revolution by swinging a counterweight from a pivot. If the center of gravity of the counterweight is 6 in. from the center of the crankshaft the required simple pendulum length is about $\frac{3}{8}$ in. If a compound pendulum is used, its radius of gyration referred to its center of gravity must not be greater than 3/16 in. or it will not be possible to obtain the desired frequency no matter where the pivot point is located. Since the radius of gyration of the counterweight about its center of gravity is many times 3/16 in., it is not possible to use the counterweight swung from a single pivot as a compound

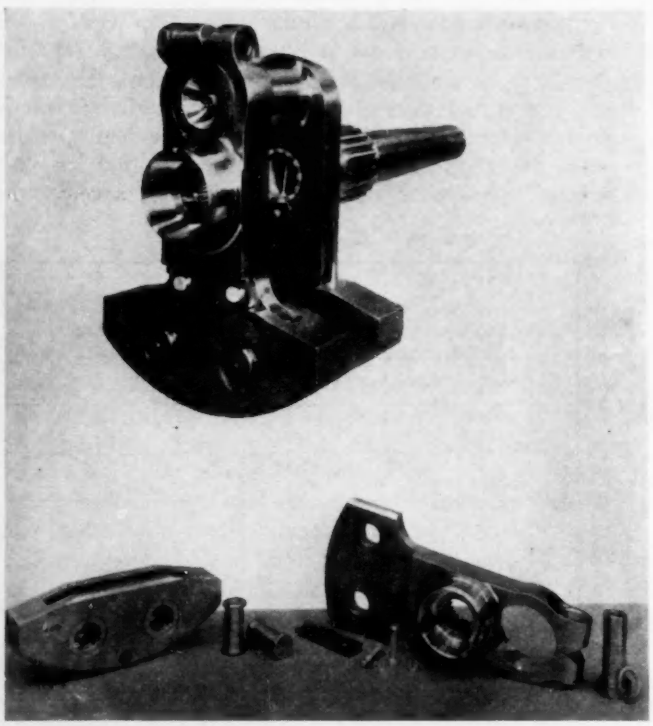

Fig. 14—Design Originated by R. Chilton of Wright Aeronautical Corp.

pendulum. Nor is it possible to use a part of the counterweight, as any mass which can be put in a crankcase and which has a radius of gyration of 3/16 in. will be hopelessly small for the purpose. Even if it were possible to obtain the desired frequency by swinging a mass from a pivot, a bearing subjected to the necessarily high centrifugal loading would be undesirable because the device depends for its effective operation upon the absence of appreciable friction.

These difficulties were cleverly overcome in a design originated by R. Chilton of the Wright company and is illustrated in Fig. 14. The counterweight is supported upon two rollers which are loose in holes in both the counterweight and the crank cheek. A moment of thought will show that this arrangement is equivalent to suspending the weight from two links as shown diagrammatically in Fig. 15. With this suspension, all points of the counterweight are constrained to move in circular paths, and since the counterweight has parallel motion when it is displaced, it behaves as a simple pendulum of length L (Fig. 15). By the use of this arrangement, it is easily possible to obtain the desired frequency of $4\frac{1}{2}$ cycles per revolution. The use of the entire mass of the counterweight in the device provides an ample margin so that its amplitude is small even during rapid acceleration from low speeds. At the same time, the use of rollers solved the problem of friction.

The effectiveness of the device is best illustrated by a series of records taken with the Prescott Torsiograph and shown in Fig. 16. The counterweight was wedged solidly while taking the left-hand series of records, while the device was operating for the right-hand series. The vibration remaining when the counterweight was pendulous was evidently so small that it could not be detected by the torsiograph.

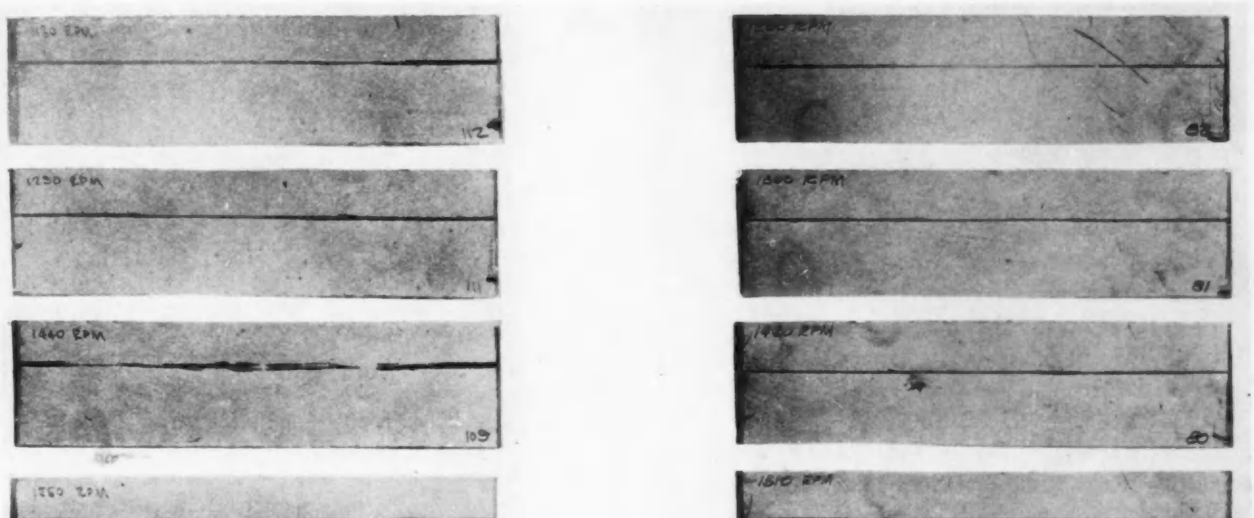

Fig. 16—Records of Torsional Oscillation of a Radial Engine Crankshaft

Relation of Exhaust Gas Composition to Air-Fuel Ratio

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Research Laboratories Section, General Motors Corp.

THE increasing use of the analyses of the exhaust gas from an operating engine to measure air-fuel ratio has made the exact relation between composition and mixture ratio of some importance. Complete analyses of the exhaust gas are slow and laborious, and a simple relation between air-fuel ratio and one or more constituents easily determined by chemical analysis, or by automatic measurement of some property of the exhaust gas, is to be preferred, provided a suitable relation or calibration is available. The data in the literature are not altogether consistent.

In the work reported in this paper, complete exhaust gas analyses for carbon dioxide, carbon monoxide, hydrogen, methane, and oxygen have been related to directly measured air-fuel ratios for three engines over a range of operating conditions and with varied air-measuring equipment. The data are consistent and they show that the exhaust gas composition is related to measured and also to computed air-fuel ratios with a deviation of about 3 per cent. The rapid determination of one or more constituents in the exhaust gas, or the measurement of some physical property, thus serves to estimate the air-fuel ratio rapidly and accurately. The results of such measurements are accordingly useful in the investigation of a variety of carburetion and fuel problems and in the rapid estimation of volumetric efficiency.

FOR a number of problems in connection with the design and maintenance of motor-vehicles, a knowledge of the fuel-air mixture being supplied to the engine under a particular set of operating conditions is of great importance. Actual measurements of ingoing air are inconvenient at best and are very difficult to carry out in moving vehicles on the road. For this reason, the determination of the air-fuel ratio being supplied to an operating engine has been made from the exhaust gas composition as determined by chemical analysis. There are available now a number of devices which measure some property of the exhaust gas such as thermal conductivity or readiness of combustion, and some that automatically determine one or more constituents by actual removal from the gas. Such properties are related to the exhaust composition and hence to the air-fuel ratio being supplied to the engine.

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The validity and usefulness of these devices, or even the usefulness of a complete chemical analysis, depend upon the relation between exhaust gas composition and air-fuel ratio. The establishment of a definite relationship between these factors provides directly a tool for determining air-fuel ratio, and such a tool has been found useful in a wide variety of problems. Thus it may be used for measurement of volumetric efficiency and air flow when the fuel rate is known.

A survey of the literature on the subject reveals that, although a variety of data on exhaust gas composition and air-fuel ratio has been published, the data of different investigators are not entirely consistent. Furthermore, no published investigation has combined the actual measurement of air-fuel ratios by means of different air-measuring devices, independently calibrated, with complete analyses of the exhaust gas from different engines under a variety of operating conditions. Accordingly, the investigation recorded in this paper was undertaken.

As a result of this work it has been concluded that there are consistent relationships between air-fuel ratio and exhaust gas composition; and the difference between the mixture ratio as computed from the exhaust analysis and the ratio as measured on the ingoing charge has averaged less than 3 per cent. Consequently, the exhaust gas composition may be used as an index of air-fuel ratio with a good degree of confidence, as may also some measurements of the physical properties or a determination of a single constituent in the exhaust gas, when properly correlated.

Data of Previous Investigators

Watson[12] reported exhaust gas compositions at various air-fuel ratios over the entire operating range of a multi-cylinder engine. He determined only the carbon dioxide, oxygen, and carbon monoxide in the exhaust gas by direct chemical analysis and calculated the hydrogen and methane contents by "Ballantyne's Rule"[2], which relates the hydrogen and methane concentrations in the exhaust gas as linear func-

tions of the carbon monoxide concentration in the following manner, $H_2 = 0.36\ CO$ and $CH_4 = 0.12\ CO$. The air-fuel ratios were determined by direct measurement of the air and fuel inputs to the engine.

Fenning[4] prepared "exhaust gas charts" showing the relation of exhaust gas composition to air-fuel ratio from data obtained without the use of an engine. Portions of mixtures of known air-fuel ratio were burned in an explosion vessel and analyses were conducted on the products of combustion. A series of exhaust gas analyses was then made on a single-cylinder engine on which the air-fuel ratio was determined by measuring the air and fuel inputs to the engine. Results seemed to agree fairly well with the explosion vessel data although only a relatively small amount of engine data was presented. Fenning concluded that very small amounts of saturated and unsaturated hydrocarbons were present in exhaust gas and that the hydrogen content of the exhaust gas from an engine operating on gasoline was given by the empirical relationship,

$$H_2 = \frac{1}{4.62\ CO^{1.38}}$$

Best[3] made direct measurements of air-fuel ratio and made Orsat analyses of the exhaust gas for carbon dioxide, oxygen, and carbon monoxide from the individual cylinders and from the exhaust pipe of a six-cylinder engine. However, attempts to relate the Orsat analyses of the exhaust gas to air-fuel ratio by several methods resulted in marked differences between the measured and computed air-fuel ratios.

More recently, Gerrish and Tessmann[7] have found, from data obtained on one multi-cylinder and on four single-cylinder engines, that for a standard grade and fighting grade aviation gasoline and two Diesel engine fuels of approximately the same hydrogen-carbon ratio as the aviation fuels, the hydrogen content of the exhaust gas may be expressed as a linear function of the carbon monoxide content by the equation, $H_2 = 0.51\ CO$. The methane content was found to be constant at 0.22 per cent, regardless of the air-fuel ratio. Experimentally determined air-fuel ratios were reported to agree with air-fuel ratios calculated from the exhaust gas analyses within $\pm$ 2 per cent, although complete data are not given.

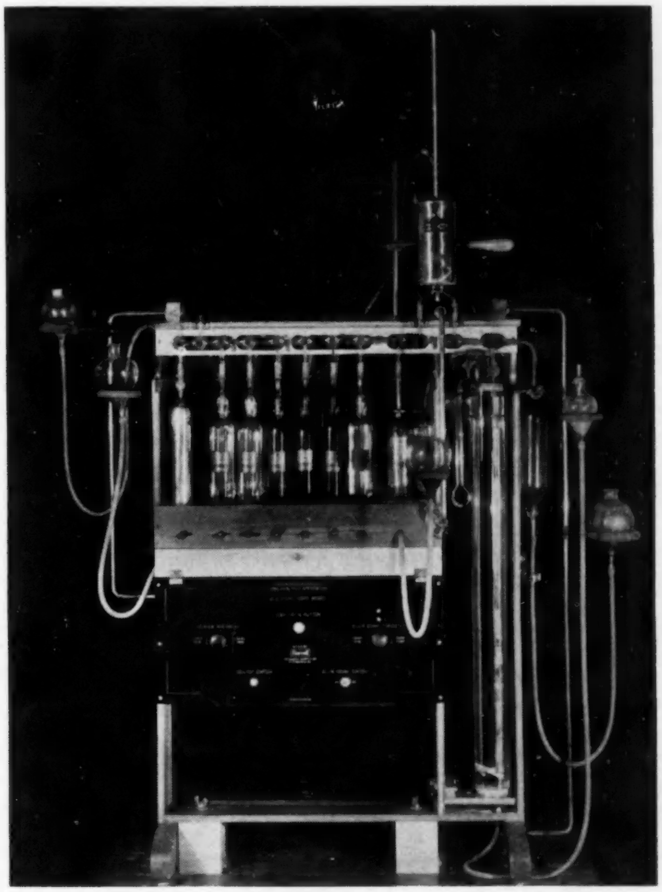

Fig. 1—Gas Analysis Apparatus

Fieldner and his co-workers[5, 6] made thorough analyses of the exhaust gas from a large number of motor-vehicles operating under various conditions. Air-fuel ratios, however, were computed from the exhaust gas analyses, and direct measurements of fuel and air are not available.

Graf, Gleeson, and Paul[8] report fair agreement between methane contents determined by analysis and those calculated from the exhaust gas analysis by means of a carbon material balance. The methane was assumed to contain all of the carbon not accounted for after the carbon in the carbon dioxide and carbon monoxide had been computed. The methane contents so obtained were found to increase with a decrease in air-fuel ratio.

Experimental Procedure and Data

The experimental portion of this study consisted of measuring the air and fuel inputs to three gasoline engines operating under various conditions of load, speed, and carburetor setting, and of analyzing samples of exhaust gas collected under these conditions. The engines used were a 1934 eight-cyl-

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inder valve-in-head automobile engine with a 3 3/32-in. bore and 4⅝-in. stroke, a 1934 six-cylinder valve-in-head automobile engine with a 3 5/16-in. bore and 4-in. stroke, and a 1935 six-cylinder L-head truck engine with a 4¼-in. bore and 4½-in. stroke. The use of the truck engine was obtained through the courtesy of the Ethyl Gasoline Corp. at whose Detroit Engineering Laboratories the engine was used. The engines were directly connected to electric dynamometers.

The air consumed by the valve-in-head engines was measured by means of an air-meter which was a motor-driven, positive-displacement Roots type blower equipped for automatic operation and especially designed for engine studies. The speed of the blower motor was automatically varied so as to maintain an almost zero pressure-differential between the intake and discharge sides of the blower. This was accomplished by means of a differential-pressure-operated diaphragm which, through contacts and relays, started, stopped, or reversed a small electric motor which varied the setting of a slide-wire rheostat in the field circuit of the blower motor. The airmeter was calibrated against a master orifice set, which had previously been calibrated against a positive-displacement tank, to read directly in cubic feet of air at atmospheric temperature and pressure.

The air input to the truck engine was determined by means of a calibrated orifice belonging to the Ethyl Gasoline Corp.

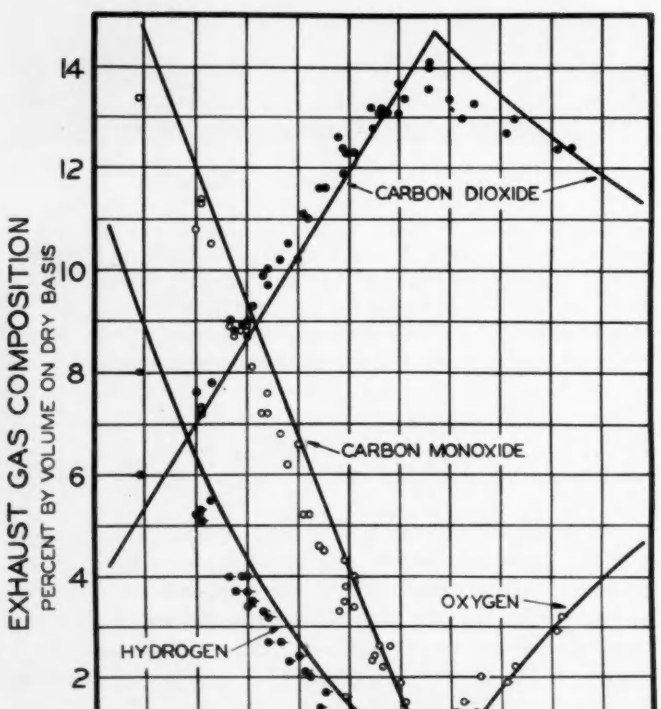


Fig. 2—Comparison of Observed Experimental Data with Computed Relationships between Exhaust Gas Composition and Air-Fuel Ratio

Points shown were obtained experimentally by metering the air and fuel charge to the engine and by analysis of the exhaust gas. Solid lines represent computed relationships based on a fuel composition corresponding to a hydrogen-carbon ratio indicated by the formula C_8H_{17}, water-gas reaction equilibrium constant of 3.8, and the formation of 0.15 mols of CH_4 per mol of C_8H_{17}.

Table 1—Fuel Inspection Data

Specific Gravity at 60 deg. fahr.: 0.733
Distillation Range:

Volume Distilled cc.	Temperature Deg. Fahr.
Initial	84
10	118
20	149
30	183
40	212
50	237
60	261
70	284
80	318
90	353
End Point	396

Recovery: 97.5 cc.
Residue: 1.1 cc.
Octane Rating: 69.

This orifice was of the rounded type with a diameter of 2.25 in. A large surge chamber was installed between the orifice and the carburetor air intake on the engine in order to minimize pulsation effects.

The volume of fuel used by the engines was measured in the fuel burettes used in making conventional dynamometer fuel consumption tests, and conversion to weights of fuel was made with suitable corrections of density for temperature.

The fuel used in this study was a commercial regular-grade gasoline. This gasoline, already treated with tetraethyl lead, was used in the two automobile engines. The tests on the truck engine were made with a fuel of the same make and grade as the other fuel except that it had not been treated with tetraethyl lead when manufactured, and 3.0 cc. of tetraethyl lead per gallon were added to this gasoline to prevent detonation. Representative inspection data on the fuel used are shown in Table 1.

Exhaust gas samples, obtained by withdrawal from the exhaust pipe close to the exhaust manifold, were collected in 250 cc. glass gas-sampling bottles by displacement of mercury. The bottles were sealed under pressure in order to prevent contamination of the sample with air in case of stopcock leakage. A Standard-Burrell gas analysis apparatus, shown in Fig. 1, was used for the analysis of the samples. The carbon dioxide and oxygen were determined in the usual manner by absorption in sodium hydroxide and alkaline pyrogallol, respectively. The carbon monoxide content was found either by slow combustion with air or oxygen over a heated platinum filament together with hydrogen and methane, or by absorption in acid cuprous chloride solution followed by cuprous sulphate-beta naphthol mixture[1]. Mercury was used as the confining liquid in the measuring burette and leveling bulbs of the gas analysis apparatus in order to minimize errors due to the absorption by the water of the carbon dioxide in the exhaust gas and from the slow combustion process. Other usual necessary precautions were taken in order to insure accurate analyses. The analyses are expressed on a dry basis since the water formed during combustion in the engine is condensed and is therefore not measured during the analysis.

By means of oxygen and nitrogen material balances and a few simple stoichiometrical calculations, the amount of water formed by combustion, the fuel composition, and the air-fuel ratio on a weight basis were computed for each test from the exhaust gas analysis. In all computations, the volumetric

composition of air was taken as 20.9 per cent oxygen and 79.1 per cent nitrogen and the molecular weight as 29.

The experimental data together with results computed from these data are shown in Table 2.

Discussion

An examination of the data has indicated that there is no grouping of the data relating the exhaust gas composition to air-fuel ratio, as far as the different engines, operating conditions, or air-measuring procedure are concerned. In other words, the exhaust composition seems to be related to air-fuel ratio and not to engine, load, or speed. This would not be so, however, if the engine were missing on one or more cylinders, or otherwise not operating properly. It is also of interest to note that no significant difference was found between the two air-metering devices which were independently calibrated.

A graphical representation of all of the data obtained is shown in Fig. 2, in which the per cent by volume of the different constituents is plotted against the measured air-fuel ratio. These percentages are percentages of the dry gas, or as ordinarily given by the usual gas analysis procedures. The data appear quite consistent. The solid lines on the chart represent the composition which may be computed as shown

Table 2—Observed and Computed Data

			Exhaust Gas Composition					Air Fuel Ratio				
Test Number	Engine Speed R.P.M.	Load	CO_2 Per Cent	O_2 Per Cent	CO Per Cent	H_2 Per Cent	CH_4 Per Cent	Measured	Computed from Exhaust Gas Analysis	From Charts in Fig. 2	Computed Fuel Composition—Atoms Hydrogen per Atom Carbon	Value of $\frac{CO \times H_2O}{CO_2 \times H_2}$
Eight-Cylinder Valve-in-Head Engine												
1	1000	Full	9.9	0.0	7.2	3.3	0.3	11.3	11.9	11.8	2.00	2.98
2	"	"	9.0	0.1	8.9	4.0	0.4	10.6	11.1	11.2	2.04	3.44
3	"	"	13.2	0.3	2.2	0.4	0.3	13.6	13.9	13.9	2.04	...
4	"	"	13.0	1.5	0.8	0.0	0.2	15.2	15.6	15.2	2.19	...
5	2000	Full	12.3	0.0	4.0	0.8	0.3	13.1	13.1	13.3	1.98	...
6	"	"	9.3	0.1	8.1	3.5	0.5	11.1	11.4	11.5	2.14	3.64
7	"	"	11.0	0.0	5.2	2.0	0.4	12.2	12.6	12.5	2.36	3.73
8	"	"	7.8	0.2	10.5	5.5	0.2	10.3	10.7	10.5	2.10	3.30
9	"	"	13.1	0.1	2.6	0.7	0.1	13.8	13.7	13.7	2.01	4.26
10	2000	Road	12.6	0.0	3.3	1.2	0.1	12.8	13.4	13.3	2.12	3.34
Six-Cylinder L-Head Engine												
1	1500	Full	11.9	0.1	4.3	1.6	0.2	12.9	13.0	13.0	2.12	3.36
2	"	"	12.3	0.1	3.4	0.9	0.5	13.1	13.2	13.4	2.15	4.77
3	"	"	10.2	0.2	6.6	2.4	0.2	12.0	12.3	12.1	2.10	4.07
4	"	"	8.7	0.2	8.6	4.0	0.4	11.0	11.4	11.3	2.23	3.68
5	"	"	8.7	0.3	8.9	3.7	0.3	11.0	11.3	11.3	2.09	3.98
6	"	"	7.3	0.3	11.4	5.1	0.5	10.1	10.3	10.3	2.01	4.05
7	"	"	7.2	0.2	11.3	5.3	0.3	10.1	10.4	10.3	2.11	4.11
8	"	"	12.8	0.0	2.4	0.6	0.5	13.5	13.7	13.7	2.33	5.08
9	"	"	13.4	0.2	1.5	0.0	0.3	14.1	14.4	14.2	2.18	...
10	"	"	13.6	0.4	0.8	0.0	0.4	14.6	14.7	14.6	2.27	...
11	"	"	13.4	0.5	0.9	0.2	0.2	15.0	15.0	14.9	2.30	...
12	"	"	13.3	1.3	0.6	0.2	0.1	15.5	15.6	15.2	2.18	...
13	"	"	12.7	1.9	0.2	0.0	0.2	16.1	16.5	16.4	2.43	...
14	"	"	12.4	2.9	0.0	0.0	0.2	17.1	17.2	17.1	2.30	...
Six-Cylinder Valve-in-Head Engine												
1	1000	Full	13.0	2.2	0.2	0.0	0.1	16.2	16.4	16.4	2.15	...
2	"	"	12.4	3.2	0.1	0.2	0.0	17.2	17.4	17.3	2.14	...
3	"	"	13.1	0.7	1.9	0.5	0.2	14.0	14.3	13.9	2.05	4.27
4	"	"	13.2	1.3	0.9	0.1	0.2	15.1	15.3	15.3	2.12	...
5	"	"	11.6	0.2	4.5	1.7	0.3	12.5	12.9	12.8	2.12	3.44
6	"	"	10.0	0.3	7.2	2.7	0.7	11.4	11.5	11.9	2.02	3.68
7	2000	Full	6.0	0.2	13.4	8.0	0.5	8.9	9.3	9.4	2.13	3.41
8	"	"	8.9	0.1	8.8	4.0	0.3	10.9	11.2	11.2	2.11	3.56
9	"	"	9.0	0.2	8.8	3.4	0.4	11.0	11.2	11.3	2.02	4.08
10	"	"	10.2	0.1	6.8	2.7	0.3	11.6	12.0	12.0	2.09	3.65
11	"	"	12.3	0.2	3.8	1.2	0.2	12.9	13.2	13.2	2.00	3.78
12	"	"	13.1	0.2	2.6	1.0	0.1	13.6	13.7	13.6	2.00	2.96
13	"	"	14.0	0.7	0.5	0.0	0.1	14.6	15.1	14.6	2.07	...
14	2000	Road	13.3	2.0	0.2	0.0	0.1	15.6	16.2	16.1	2.06	...
15	"	"	13.8	0.8	0.6	0.0	0.1	14.2	15.2	14.6	2.10	...
16	"	"	13.2	0.2	2.3	0.8	0.1	13.4	13.9	13.7	2.05	3.26
17	"	"	11.6	0.3	4.6	1.4	0.4	12.4	12.8	12.9	2.05	4.19
18	"	"	10.5	0.3	6.2	2.3	0.4	11.8	12.2	12.2	2.07	3.71
19	"	"	8.8	0.2	8.7	3.7	0.5	10.7	11.2	11.3	2.20	3.90
20	"	"	7.6	0.5	10.8	5.2	0.2	10.0	10.6	10.5	2.00	3.55
21	1000	Road	9.7	0.3	7.6	3.2	0.5	11.4	11.5	11.7	2.04	3.43
22	"	"	11.1	0.2	5.2	2.1	0.4	12.1	12.5	12.5	2.15	3.35
23	"	"	12.4	0.2	3.5	0.9	0.5	12.9	13.1	13.4	2.05	4.54
24	"	"	13.7	0.3	1.2	0.1	0.4	14.0	14.2	14.2	2.13	...
25	"	"	14.1	0.6	0.5	0.0	0.3	14.6	14.8	14.6	2.05	...
Average	"	"					0.3	12.9	13.2	13.2	2.12	3.77

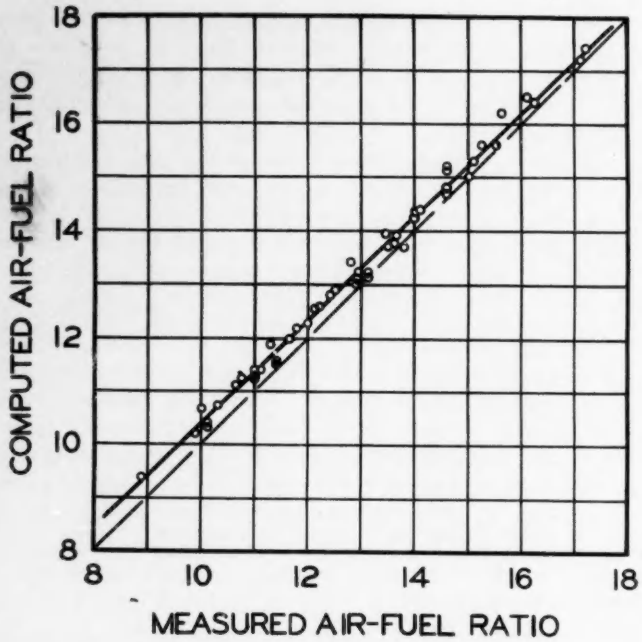


Fig. 3—Comparison of Measured and Computed Air-Fuel Ratios

Measured air-fuel ratios obtained experimentally by metering the air and fuel charge to the engine. Computed air-fuel ratios computed directly from analysis of exhaust gas obtained when air and fuel charge was measured.

in the Appendix on the assumptions that the composition of the fuel is such as to correspond to 84.9 per cent carbon and 15.1 per cent hydrogen, that the relation between the constituents in the gas is such that:

$$\frac{CO \times H_2O}{CO_2 \times H_2} = 3.8$$

and that 1.875 per cent of the carbon in the fuel appears as methane, CH_4, in the exhaust. There is also, of course, the additional assumption that the air and fuel entering the engine come out in the exhaust without any selective diversion of any constituent. These assumptions seem reasonably in accord with known facts and provide a definite basis for computing the exhaust composition from fuels of different elementary composition.

The results of such computations, further discussion and details of which are given in the Appendix, seem, as shown by the lines on the chart, to be quite closely in accord with the direct experimental data. The deviations between the observed and computed compositions are of about the same magnitude as the probable experimental error.

In addition to comparing the calculated exhaust gas composition with the observed, it is also possible, from the experimentally determined exhaust gas composition, to compute the corresponding air-fuel ratio, assuming only that there has been no selective diversion of any constituent in the engine. Details of such computations are given in the Appendix, and the results of such calculations are represented graphically in Fig. 3. There, the directly measured air-fuel ratio is plotted against the air-fuel ratio computed from the observed exhaust gas composition. The correlation appears quite consistent and the computed ratio is slightly greater than the observed by about 3 per cent on the average. The cause of this discrepancy is not known with certainty, but, considering even the difficulty of measuring large volumes of air to an accuracy of 3 per cent, this slight difference has not at present seemed of major importance.

A comparison of the air-fuel ratios computed directly from the exhaust gas analyses with those derived from the exhaust gas analyses by use of the computed relations in Fig. 2 is shown in Fig. 4. These data indicate an agreement well within the limits of experimental error. No deviations significantly consistent, except those due to the imperfect fuel distribution referred to in the Appendix in the air-fuel ratio range from 14 to 16, are apparent.

Consequently, it is reasonable to regard the data on exhaust gas composition as a function of air-fuel ratio, given in Fig. 2, as quite reliable. The data were obtained with different engines, operating conditions, and air-measuring equipment. The air-fuel ratio computed from the observed composition agrees quite well with the measured air-fuel ratio. There is good agreement between the observed exhaust gas composition and that computed with assumptions involving the composition of the fuel.

Such a chart as Fig. 2 has been found to be useful in a number of ways. A measurement of one or more constituents in the exhaust gas instead of a complete analysis serves to determine the air-fuel ratio, provided the engine is operating normally. Such data have been used to calibrate various exhaust gas testers which depend upon measuring one or more constituents or some property of the exhaust gas. A quick and accurate means of measuring air-fuel ratio has been found to have many applications in development work as well as in maintenance of automobiles.

Appendix

When a hydrocarbon such as gasoline is burned in an engine under ideal conditions with less than the theoretical amount of air required for complete combustion, the following primary products of combustion are obtained: carbon dioxide, carbon monoxide, water, and hydrogen. For a given set of temperature and pressure conditions, an apparent equilibrium is reached among the four products of combustion, as indicated by the following equation which is known as the water-gas reaction:

$$CO_2 + H_2 \rightleftarrows CO + H_2O \quad (1)$$

The extent to which this reaction proceeds in either direction is expressed by the water-gas reaction equilibrium constant K which is defined as the product of the concentrations of CO and H_2O divided by the product of the concentrations of CO_2 and H_2, or

$$K = \frac{CO \times H_2O}{CO_2 \times H_2} \quad (2)$$

All but one of the four constituents required to compute K are obtained directly from exhaust gas analyses. Water, the other constituent, which condenses out of the exhaust gas before analysis, is computed as already stated. The apparent equilibrium constants shown in Table 2 were computed in this manner. The accuracy with which the equilibrium constant can be ascertained depends on the accuracy of the exhaust gas analysis. Hydrogen is the most difficult constituent to determine, as it occurs in much smaller concentrations than the other constituents, especially at air-fuel ratios near the theoretical for complete combustion. Thus, at low hydrogen concentrations, a small error in determining the hydrogen will have a very large effect on the value of the equilibrium constant. For this reason, equilibrium constants are not shown in Table 2 for exhaust gas analyses showing hydrogen

contents of 0.5 per cent or less. The average value of the equilibrium constants shown in Table 2 is 3.77, or about 3.8. Lovell and Boyd[9] by a mathematical analysis of the data of Fieldner and his associates[5, 6], found values ranging, for the most part, between 3.0 to 4.0 with an average value of 3.6.

In addition to the four constituents already mentioned, actual exhaust gas analyses show the presence of an average of 0.3 per cent methane in the products of combustion of gasoline. Although reported as methane, the small amount of hydrocarbon found in the exhaust gas makes it very difficult to be certain that it is all methane and not a mixture of hydrocarbons. The reason for the presence of methane in the exhaust gas is not apparent. Methane is very stable thermally and as a decomposition product of gasoline may possibly be formed during some portion of the combustion cycle under conditions which do not favor its combustion.

From the complete analysis of the exhaust gas, the fuel composition was determined in terms of the ratio of the hydrogen to carbon atoms. The average hydrogen-carbon ratio of the fuel according to Table 2 is 2.12. This composition corresponds to a fuel containing 84.9 per cent carbon and 15.1 per cent hydrogen by weight, and it agrees reasonably well with gasoline compositions reported in the literature[7, 10]. The relationships which will be derived later are strictly valid only for fuels of about this composition. The approximate composition of an unknown fuel may be determined by use of the following relationship[11], $\%H = 26 - 15d$, where H represents the hydrogen and d the specific gravity of the fuel at 60 deg. fahr.

In the calculations which follow, the formula C_8H_{17}, corresponding to the hydrogen-carbon ratio 2.12, is used to express the fuel composition. It is not intended to imply that the molecular weight of the fuel is necessarily that corre-

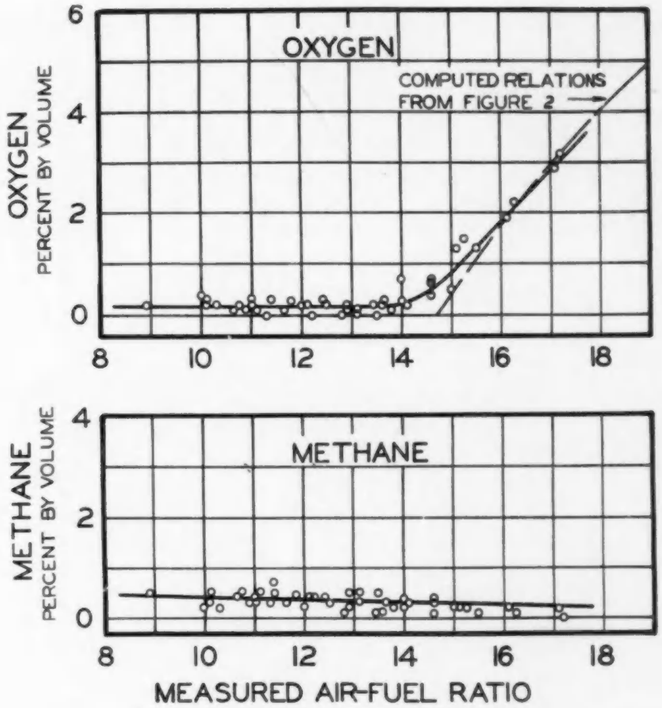


Fig. 5—Relation of Observed Oxygen and Methane Contents of Exhaust Gas to Air-Fuel Ratio

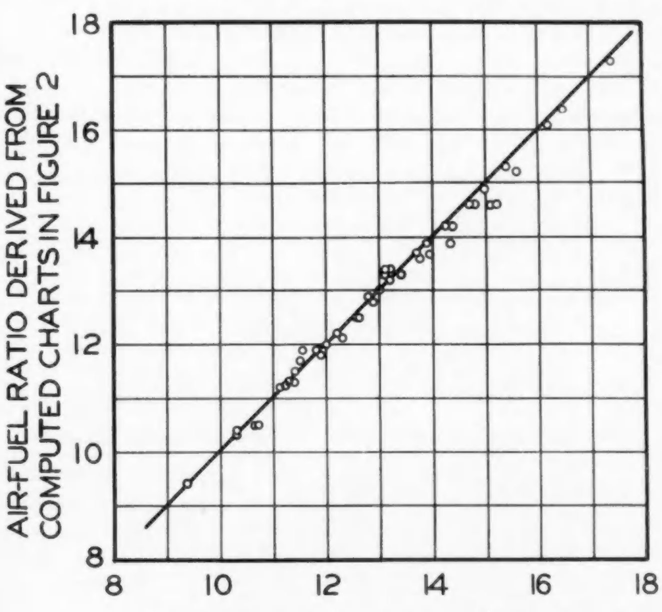


Fig. 4—Comparison of Air-Fuel Ratios Computed Directly from Exhaust Gas Analyses with Air-Fuel Ratios Derived from Charts in Fig. 2

Computed air-fuel ratios obtained by direct calculation from observed exhaust gas analyses. Derived air-fuel ratios obtained from observed exhaust gas analyses by use of computed charts in Fig. 2. Each point shown is the average of the air-fuel ratios indicated by the charts for the individual constituents in the exhaust gas.

sponding to this formula, and the formula C_8H_{17} is used in this work merely as a means of indicating the hydrogen-carbon ratio in a manner more convenient to use in the following stoichiometrical calculations.

Since the average values of the equilibrium constant, the fuel composition, and the methane content have been determined experimentally, it is possible from these data to compute the relationships existing between exhaust gas composition and air-fuel ratio. The equation for the combustion of one mol of fuel of composition C_8H_{17} with various proportions of air is:

$$C_8H_{17} + (n)\,O_2 + \left(\frac{79.1n}{20.9}\right) N_2 = (a)\,CO_2 + (8-a-p)\,CO + (b)\,H_2O + (8.5-b-2p)\,H_2 + (p)\,CH_4 + \left(\frac{79.1n}{20.9}\right) N_2 \qquad (3)$$

The sum of the terms $(n)\ O_2$ and $\left(\frac{79.1n}{20.9}\right) N_2$ represents the number of mols of air per mol of fuel used for the combustion. The coefficients of the other terms in equation (3) indicate the number of mols of the various constituents formed per mol of fuel burned. The term $\left(\frac{79.1n}{20.9}\right) N_2$ appears on both sides of the equation because the nitrogen is assumed to pass through the combustion process without reaction.

It was found by trial that the methane coefficient (p) should have a value of 0.15 in order to give a methane content of about 0.3 per cent in the dry exhaust gas. This corresponds to the conversion of 1.875 per cent of the carbon in the fuel to methane. After substituting this value of (p) in equation (3), a material balance for the oxygen shows that

$$n = a + \left(\frac{7.85-a}{2}\right) + \frac{b}{2} \qquad (4)$$

By substituting in equation (2) the average experimental

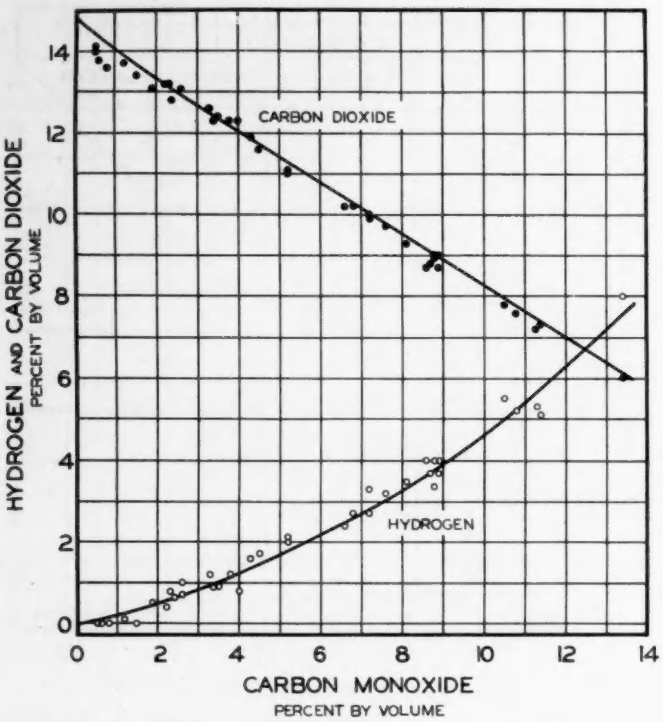


Fig. 6—Relation of Carbon Monoxide to the Carbon Dioxide and Hydrogen Contents of Exhaust Gas
Points shown obtained from observed exhaust gas analyses. Lines represent computed relationships taken from Fig. 2.

value of the equilibrium constant and the values of the other constituents from equation (3), this equation becomes

$$K = \frac{CO \times H_2O}{CO_2 \times H_2} = \frac{(7.85-a)\ (b)}{(a)\ (8.2-b)} = 3.8 \qquad (5)$$

Exhaust gas compositions may now be computed by solving equations (4) and (5) simultaneously at various values of (n). Air-fuel ratios are computed by dividing the weight of air (given by the sum of the terms (n) O_2 and $\left(\frac{79.1n}{20.9}\right)N_2$) by the weight of fuel (given by the term C_8H_{17}). When (n) equals 11.95, the terms (7.85-a) and (8.2-b) become zero and consequently the carbon monoxide and hydrogen contents become zero. This indicates that just enough air is being supplied to the fuel to burn it completely to carbon dioxide and water (except for the small amount of methane which remains unburned). The air-fuel ratio of 14.7 corresponding to this value of (n) is known as the theoretical ratio for complete combustion of the fuel being considered. Air-fuel ratios below 14.7 are said to be richer and those above this figure are said to be leaner than the theoretical.

The relationships computed as described above are indicated as the solid lines in Fig. 2, which shows the exhaust gas composition expressed in per cent by volume on a dry basis as a function of the air-fuel ratio on a weight basis. In the vicinity of the air-fuel ratio for theoretically complete combustion, i.e., from about 14 to 16 air-fuel ratio, the observed carbon dioxide contents lie below the computed values and, at the same time, carbon monoxide and hydrogen are present in the exhaust gas at air-fuel ratios beyond 14.7. These discrepancies are due principally to imperfect fuel distribution which causes some cylinders to operate leaner than 14.7 air-fuel ratio and others to operate richer than this figure. On account of this imperfection of distribution, carbon monoxide, hydrogen, and oxygen are found together in the exhaust gas.

The observed exhaust gas compositions also indicate the presence of a small amount of oxygen at air-fuel ratios below 14.7. This may be seen more clearly in upper half of Fig. 5, which shows the observed oxygen content as a function of the measured air-fuel ratio. The broken line represents the computed relationship taken from Fig. 2. Between 8 and about 14 air-fuel ratio, the oxygen content averages about 0.2 per cent. It then starts to increase as the air-fuel ratio is increased. Due to the absence of complete equality of fuel distribution, in practice, the observed oxygen content is higher than the computed between 14 and 16 air-fuel ratio. Beyond this range, however, the observed oxygen contents seem to agree with the computed.

The relation of the observed methane content to the measured air-fuel ratio is shown in the lower portion of Fig. 5. The methane content of the exhaust gas varies between 0 and 0.7 per cent and appears to decrease slightly as the air-fuel ratio is increased. The average value of the methane concentration is 0.3 per cent according to Table 2. This figure compares favorably with the average of 0.22 per cent reported by Gerrish and Tessmann[7] who found the methane content of exhaust gas to be independent of the air-fuel ratio.

A comparison between the observed and computed exhaust gas compositions is shown in Fig. 6. In this figure, the points shown represent observed carbon dioxide and hydrogen contents plotted as a function of the observed carbon monoxide concentrations. The solid lines indicate the computed relationships as taken from Fig. 2. The agreement between the observed hydrogen data and the computed hydrogen line is very good with the observed data falling on both sides of the computed relationship. The agreement between the observed and computed carbon dioxide contents is not quite as good, although deviations are in the order of only 0.1 to 0.3 per cent carbon dioxide. The slightly greater deviations in carbon dioxide content observed between 0 and 2 per cent carbon monoxide are undoubtedly due to the effects of the imperfect fuel distribution already mentioned. The agreement between the observed and computed exhaust gas compositions indicates that the bases of the computed relationship, i. e., the values of the water-gas reaction equilibrium constant, fuel composition, and methane content, appear to be very reasonable.

Discussion

—**Dr. H. C. Dickinson**
National Bureau of Standards

THERE never has been very much doubt about the reasonably definite relationship between the composition of exhaust gases and mixture ratio. It is fortunate, however, to have available in convenient form the results of a careful series of observations which serve to set the practical limits within which complete or partial measurements of exhaust gas composition can be used to indicate air fuel ratio of the initial mixture.

The uncertainties due to irregular distribution in multicylinder engines and to possible occasional misfire are necessarily inherent in the problem and must always introduce additional uncertainty. The experimental work of the authors, including three different engines of radically different types, indicates to what extent these uncertainties may be neglected under favorable operating conditions.

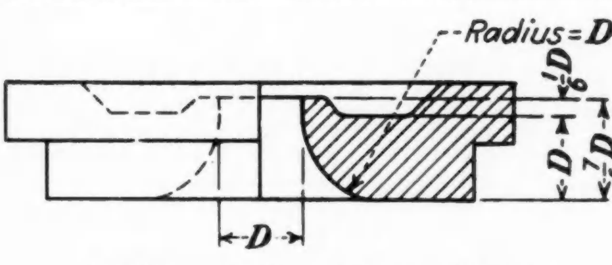
The reader should be warned, however, that results of the same degree of reliability can be expected only when adequate care is exercised to secure uniformity of fuel distribution and consistent firing on all cylinders.

—Earl Bartholomew
Ethyl Gasoline Corp.

THE problem of air-fuel ratio measurements is divided, by the equipment which may be utilized, into determinations made in the laboratory and those made on the road. In the former case it is usually possible to install on the engine the necessary instrumentation for the volumetric measurement of air and fuel although carburetor characteristics may make the installation difficult, the volume of experimental work to be done may not warrant it and the time available may preclude it. Exhaust gas analysis then becomes the only alternative.

For those who may be interested in the round-edge orifices mentioned in the paper in connection with the volumetric air measurements on the L-head truck engine, the following information may be of interest.

All orifices were machined in accordance with the general design described in Bulletin No. 207 of the University of Illinois Engineering Experiment Station and shown in Fig. *A*. All dimensions are expressed in terms of throat diameter. The flow coefficients shown in Fig. *A* were determined by calibration against gas meters supplied by the Detroit City Gas Co. which were in turn calibrated against actual volume displacement immediately before and after the orifice calibrations. The coefficients for the larger sizes approach unity quite closely. In conjunction with burettes for the volumetric measurement of fuel the orifices provide accurate instrumentation at low cost. Fig. *B* shows a range of sizes of the above described **orifices.**



Orifices No. 1 to 5 are similar to above sketch and are attached to surge tank by clamping to a threaded adapter. Orifices No. 6 to 12 are threaded.

Orifice No.	Throat Diameter, D, In.	Coefficient at Pressure Differential of 0.5 In. Water	Coefficient at Pressure Differential of 3.0 In. Water	Lb. of Air per Hr. 29.92 In. Hg., Dry Air, 60 deg. fahr. at Pressure Differential of 0.5 In. Water	Lb. of Air per Hr. at Pressure Differential of 3.0 In. Water
1	0.250	0.917	0.939	3.8	9.8
2	0.328	0.966	0.972	7.2	17.7
3	0.430	0.967	0.970	12.0	29.6
4	0.547	0.972	0.980	19.5	48.5
5	0.719	0.987	0.989	34.0	85.0
6	1.000	0.990	0.988	68.0	163.0
7	1.250	0.985	0.990	106.0	258.0
8	1.500	0.987	0.992	150.0	370.0
9	1.800	0.991	0.992	220.0	535.0
10	2.250	0.989	0.995	340.0	840.0
11	2.750	0.991	0.993	510.0	1250.0
12	3.500	0.993	0.996	1050.0	2030.0

Fig. *A*—(Bartholomew Discussion) General Design for Orifices Made in Accordance with Design Described in Bulletin No. 207 of the University of Illinois Engineering Experiment Station

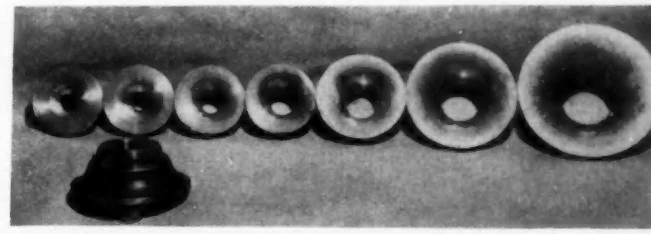


Fig. *B*—(Bartholomew Discussion) A Range of Sizes of Orifices Made in Accordance with Design Shown in Fig. *A*

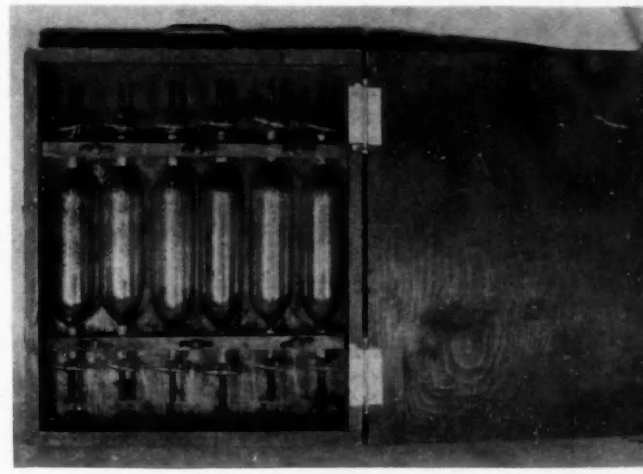


Fig. *C*—(Bartholomew Discussion) Case and Six Bottles for Taking Gas Samples on the Road

The movement of the air about a vehicle on the road causes rapid fluctuations in the pressure ahead of orifices for air measurement and the pitching of the vehicle prevents the maintenance of a vertical position for the manometer. The difficulty of obtaining across orifices or air meters the pressure drop characteristics of air cleaners, and the temperature changes introduced by moving the source of air to a position outside the hood, may be additional objections.

The routine in connection with the sampling and analysis of exhaust gas samples from vehicles on the road may be simplified and accuracy insured through the adoption of certain precautions and apparatus. Samples should be withdrawn from a point in the exhaust pipe ahead of any possible air infiltration.

Portable apparatus for gas analysis may have certain limitations but of even greater importance are the inconvenience of aspirating a sample of gas directly into an instrument and the inaccuracy introduced by the conditions which usually surround an analysis by the roadside. We have therefore adopted the practice of having the analysis apparatus permanently set up in the laboratory. Sample bottles are carried on the road in cases (Fig. *C*) which accommodate six and permit their easy insertion and removal. The hinged cover may be removed while samples are collected but protects the bottles during transit. Bottles bear permanent numbers for the later identification of samples in the laboratory. Experience has indicated that the sample bottles are self scavenging without liquid displacement when held in a vertical position with connection at the top and exhaust through the bottom cock.

Orsat apparatus which permits rapid analysis is not only an investment in time saving but also in accuracy. To pre-

vent gas absorption, only mercury or certain aqueous solutions should be used as displacement liquids. Unless it is definitely known in advance whether the sample under analysis is richer or leaner than the theoretical mixture, a determination of carbon dioxide, oxygen and carbon monoxide and subsequent reference to the chart (Fig. *B*) presented in the paper are essential in the analysis of a sample resulting from the combustion of a fuel having an average empirical formula of approximately C_8H_{17}.

Assuming a water gas constant of 3.5, our own laboratories have constructed similar charts for fuels varying from highly volatile stabilized natural gasoline to tractor distillate with an average composition of $C_8H_{14.08}$, the curves for the latter fuel being shown in Fig. *D*.

The efficiency of distribution determines the closeness of approach of the percentages of CO_2, O_2 and CO to the values shown by the calculated curves at the theoretical mixture ratio. For example, if one-half of the cylinders operate at 16 to 1 ratio and the remainder at 14 to 1, the average ratio is 15 but both CO and O_2 would be present and the CO_2 concentration would be substantially below that shown on the chart for a 15 to 1 ratio. Under such conditions a determination of the average ratio is impossible except after a complete gas analysis.

Dr. H. A. Beatty of the Ethyl Gasoline Corp. laboratories has proposed a procedure which, although not yet applied in practice, is simple and thermodynamically sound and gives results unaffected by distribution efficiency. It is described for the consideration and investigation of members interested in the problem.

Briefly stated the method consists of determining the ratio of the mols of nitrogen to the mols of carbon in the exhaust gas sample by completing the combustion in the slow combustion pipette in the presence of added pure oxygen and measuring the volume of CO_2 and N_2 in the completely burned sample. A determination of the carbon/hydrogen ratio for the fuel under consideration must also be made. The relationship between air-fuel ratio and the molar concentrations of carbon and nitrogen in the exhaust gas is derived as follows:

$$\text{Air-fuel ratio} = \text{weight air} \div \text{weight fuel} = \frac{\text{weight } N_2 \text{ in air used}}{\text{per cent } N_2 \text{ by weight in air}} \times \frac{\text{per cent } C \text{ by weight in fuel}}{\text{weight } C \text{ in fuel used}}$$

$$= \left(\frac{\text{per cent } C \text{ by weight in fuel}}{\text{per cent } N_2 \text{ by weight in air}} \times \frac{\text{mol weight } N_2}{\text{mol weight } C}\right) \frac{\text{mols } N_2}{\text{mols } C}$$

$$= (\text{per cent } C \text{ by weight in fuel} \times K) \frac{\text{Cc. } N_2}{\text{Cc. } CO_2}$$

$$= 0.0304 \times \text{per cent } C \text{ by weight in fuel} \times \frac{\text{Cc. } N_2}{\text{Cc. } CO_2}$$

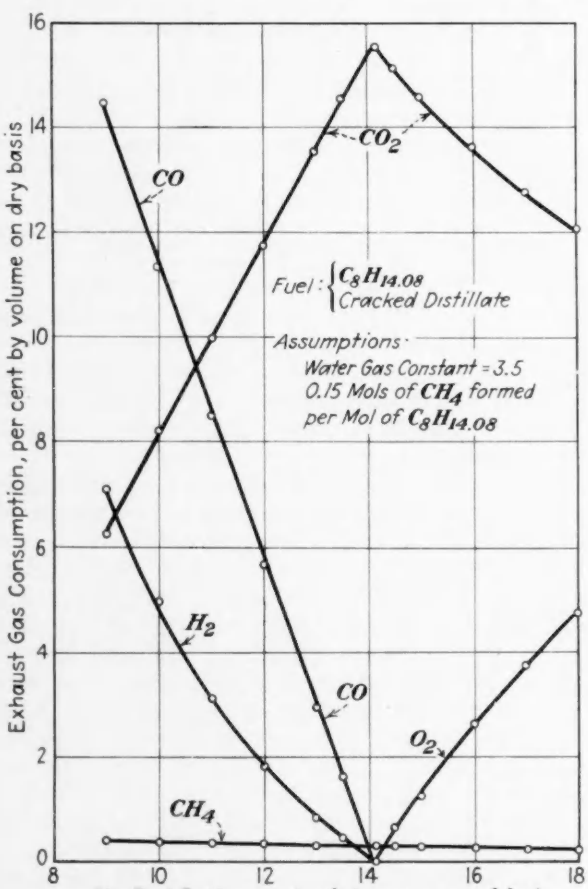


Fig. *D*—(Bartholomew Discussion) Computed Relationship of Exhaust Gas Composition to Air-Fuel Ratio for a Tractor Distillate

In connection with studies of fuel distribution by manifolds, knocking tendency of individual cylinders and similar problems, it is desirable to know the air-fuel ratio for each cylinder. In our own laboratories we have on several occasions constructed new exhaust manifolds having branches several feet in length to prevent the dilution of the exhaust from one cylinder by that from another at the points of sampling but usually such a reconstruction of the exhaust manifold causes changes in the distribution characteristics of the intake manifold, thus destroying that which it is desired to measure. Harold Chalk and Cleveland Walcutt of our laboratories have proposed the sampling of the exhaust from individual cylinders by tubes terminating near the exhaust valves and connected to evacuated sampling bottles through solenoid valves which would be opened only when the corresponding exhaust valve is open, the operation of the valves to be controlled by proper commutator mechanism synchronized with the engine crankshaft.

—Harold C. Gerrish and Fred Voss

National Advisory Committee for Aeronautics

THE authors' mathematical determination of the composition of the exhaust from the composition of the fuel, the apparent equilibrium constant, and the percentage of methane in the exhaust is a definite contribution to our knowledge of exhaust gases from internal-combustion engines. The precision of the determination of the components of the exhaust gas are especially good.

In the development of the theoretical analysis the authors have made three "assumptions": (1) the fuel to have a percentage composition equal to that found from analysis of the exhaust gases; (2) the apparent equilibrium constant to equal the average of those determined from exhaust gas analysis; and (3) the products of combustion to have a definite quantity of methane present.

The percentage of the various constituents of combustion has been calculated for the range of air-fuel ratios over which

(Continued on page 116)

Spring, Tire and Shock Absorber Testing Development

By S. Ward Widney
Director of Engineering, B. & J. Auto Spring Co.

UNTIL rather recently, this paper states, very little scientific knowledge existed in the service field with reference to the suspension—tires, springs and shock absorbers. The tires are the only element of which the operator has a fairly clear understanding; therefore they are the only element which receives intelligent attention. The shock absorbers receive some attention but not a great deal.

It is conceded that the motor-truck operators, large and small, know less about the suspension system than they do about any other important unit of the vehicle and that the reason for such lack of knowledge, is because no one knew much about it. Therefore, a little over three years ago, the author's organization decided it would endeavor to scientifically determine just what the possibilities were toward producing greater cushion in springs. Also to determine just what part the springs play in the suspension, and what part the tires and shock absorbers play.

Mr. Widney's paper shows just what happens to the three units of the suspension—tires, springs and shock absorbers on various vehicles, passenger cars, light, medium and heavy trucks, buses and taxi cabs. He shows that the suspension directly influences all maintenance costs, depreciation and accidents, as well as comfort, and that springs, tires and shock absorbers when new, vary as much as 60 per cent in their effectiveness on different vehicles. Also, that these three units depreciate in their effectiveness as a whole or separately, as much as 60 per cent, after a given period of service. Therefore, the suspension should have the same serious attention in service as motors, carburetors, ignition, etc.

[This paper was presented at the Regional Transportation and Maintenance Meeting of the Society, Newark, N. J., Oct. 31, 1935.]

FOR many years engineers have been striving toward a clearer understanding of the three agencies, tires, springs and shock absorbers, whose main function it is to absorb road shocks. Various means have been devised in the way of testing equipment, but none has produced entirely satisfactory results.

Until rather recently, engineers were compelled to be satisfied with knowledge based largely upon the static phase of the problem, in other words, on bench tests. There existed no means which would disclose to them the true dynamic picture. Many tests have been carried on covering tires, springs and shock absorbers individually, but none to my knowledge, which showed the complete reaction produced by a given combination of tire, spring and shock absorber at each of the four corners of the vehicle, resulting from varied road shocks against the tire.

The problem of road shock absorption is solely one of dynamics--not of statics. Statics concerns the load carrying phase only of the tires and springs. Therefore, the problem, if it is to be solved, must be approached from the dynamic standpoint. Moreover, the combined action of tire, spring and shock absorber must be studied. To study each separately does not go far enough, since a given tire may possess superior qualities of its own but when working with a given spring and shock absorber, the combined effect may be less desirable than if a tire of different construction or qualities were used. This may sound contrary to basic principles. However, we have proved it to be so. Much the same holds with the spring and shock absorber. Thus we must investigate not only the individual dynamic qualities of tires, springs and shock absorbers, but of greater importance, we must ascertain their combined effect.

Comfort and Safety

If the problem of the comfort and the safety of a given vehicle is to be analyzed and standards set up so as to definitely compare the performance of one vehicle with another, the action of the wheels and body must be studied separately, at each of the four corners, under the varied impacts which the tires receive when rolling over the road. Only three dynamic factors are involved; namely, the amplitudes, the velocities and the accelerations of the movements. We have established methods and means for measuring these factors incident to road shocks of large and small intensities, such as are encountered in normal operation.

Rideability and Roadability

In most instances, when we improve a vehicle's rideability (riding quality), we also improve its roadability (ability to

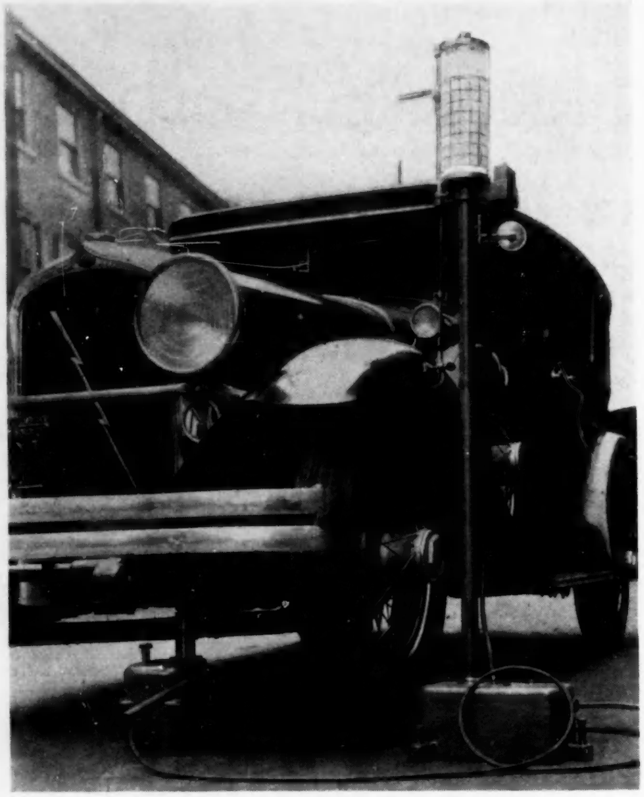


Fig. 1—The Ride-O-Graph

This is a testing device which draws a chart of the actual movements of the body and the wheels of any vehicle incident to the tire receiving a blow such as it receives when it encounters uneven road surfaces.

hold the road). This fact led to the slogan "A comfortable car is a safe car". The rideability problem is perplexing because so many factors such as tires, wheel action, unsprung weight, chassis springs, shock absorbers, shackles, body weight, body action, center of gravity, rigidity of chassis and wheel base are involved. Aside from the vehicle itself, the character of road over which it is driven, its speed and its loads also vary. Some vehicles ride comfortably at a certain speed and not so good at another speed. Some ride comfortably over a certain character of road and not so good over another kind of road. Others ride well when fully loaded but poorly with a light load. In passenger vehicles, additional factors such as the design and construction of the seats and their position in relation to the axle affect rideability. No other problem in automobile construction approaches in complexity that surrounding rideability.

Underlying Factors

Tires, springs and shock absorbers have the greatest influence on both the rideability and the roadability. Their function is not only to cushion the vehicle against road shocks and thus provide comfort and lessen the damage to the vehicle, but also to keep the wheels in contact with the road and thereby insure maximum wheel traction, particularly at high speeds and during brake application. A comfortable vehicle is not only safer, but its cost of maintenance is also lower compared with a less comfortable vehicle.

Ten years ago there was no reliable means of telling how efficiently one shock absorber on the car performed as compared with another and the same condition existed with reference to chassis springs and tires. Since that time, considerable development in testing equipment for tires, springs and shock absorbers has taken place and progress has been made. Accelerometers, inertia devices, seismographs and other equipment have been used and the information thus gained has been helpful but has not been sufficiently reliable or intelligible to result in a standard test acceptable to the industry.

With a device called the Ride-O-Graph, however, we have made progress toward establishing a definite and reliable test which shows the comparative effectiveness of tires, springs and shock absorbers in a way which we trust will be acceptable to the industry. The Ride-O-Graph was introduced about five years ago and several of the large automobile manufacturers and most of the shock absorber plants installed it in their laboratories. The B. & J. Auto Spring Co. installed it in its laboratory and has since placed others in all of its key service plants.

There are many problems in springs yet to be solved but we are greatly pleased with the progress thus far made and with the standards we have been able to establish. We have proved them thoroughly in actual practice as well as in the laboratory. The curves produced by this device are simple space-time curves. Because of their simplicity, many engineers at first were skeptical of their value but we now know how to interpret these curves quickly and accurately. The information thus obtained is reliable and presents a true picture of the comparative effectiveness of springs, shock absorbers and tires as they affect the rideability and roadability of the vehicle and influence its cost of maintenance. This has come about because we have learned to evaluate the effects of the acceleration, the velocity and the amplitude of the motions produced with various springs, shock absorbers and tires. We know that highly accelerated motions are detrimental to rideability and maintenance. They are not only discomforting to the passengers but are damaging to the vehicle in direct proportion to their rate. This must be so since acceleration $\times$ mass $=$ force, and since high accelerations result in large forces, these large forces certainly tend to damage the vehicle and increase discomfort.

The layman may not clearly understand the difference between velocity and acceleration, but he does realize that a sudden change in speed, resulting from sudden throttle opening or sudden brake application, results in forces that are uncomfortable and are hard on the vehicle. In springs, tires and shock absorbers the situation is the same, except that rapid and frequent vertical instead of horizontal accelerations produce discomfort and often damaging forces, as a result of road irregularities.

The Ride-O-Graph (Fig. 1) is a testing device which draws a chart of the actual movements of the body and the wheels of any vehicle incident to the tire receiving a blow such as it

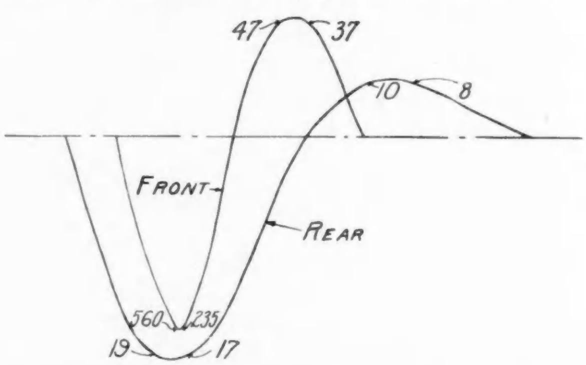


Fig. 2—Typical Ride-O-Graph Curves of a Medium Weight Passenger Car

receives when it encounters uneven road surfaces. The mechanism itself, as well as the methods employed in its use, are very simple. The vehicle to be tested is placed on a reasonably smooth, solid floor. The recorder stand is set close to the fender at the wheel to be tested. A pencil is then attached to the fender by a clamp provided for the purpose. A graduated graph sheet is attached to the recording drum and the pencil which draws the graph is placed in contact with the sheet. An electrically controlled jack, which is connected to and operates in synchronism with the recording unit, is placed under the axle next to the wheel to be raised. The wheel is then jacked up the desired distance, from one to four inches. In the rectangular base of the registering stand is a constant speed motor which rotates the drum at the top of the shaft. When the wheel is jacked up ready to be dropped to make the graph, an electric switch is turned and a tripping button is pressed, the jack is released instantly and the car drops unrestrained. Thus, the wheel is given a known impact which causes the tire and spring to compress and subsequently to recover. All the vertical motions of the body or wheel are recorded on the precisely timed chart. The graphs thus made show the exact extent of the vertical motions as well as the accelerations and velocities of such motions.

These definitely known impacts, such as a car may receive while traveling over the road, are delivered to each wheel in succession, and the body motions recorded. Then the recording unit is attached to the wheel and the motion of each wheel is recorded in the same manner. Thus the car automatically writes its own story, and the effectiveness of the tire, spring and shock absorber is definitely revealed. After these motions are recorded at each wheel, the reactions at each of the other three corners can be recorded separately. An analysis of the gyroscopic movements of the body can

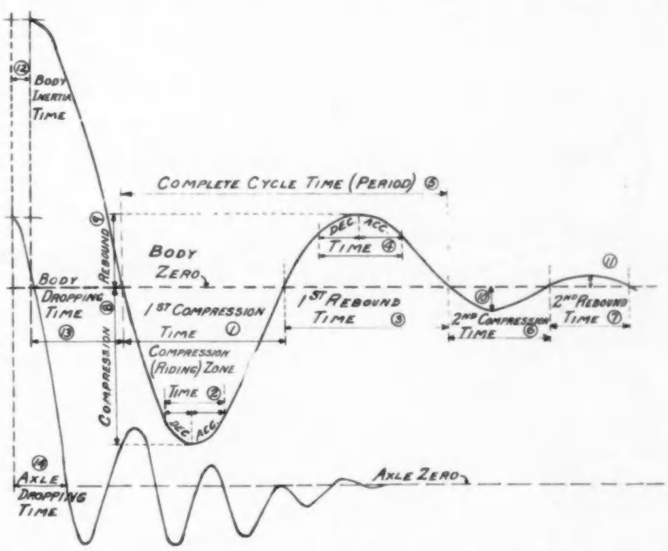


Fig. 4—Chart of the 14 Main Factors Involved in the Suspension Problem

Owner's Name	Name of Car	Model		Year		Shock Absorber	Remarks	
						DELCO		
	One Wheel Test	R. F.	L. F.	R. R.	L. R.	Tire Size	F.	R.
	Two Wheel Test	R. F.	L. F.	R. R.	L. R.	" Pressure	F.	R.

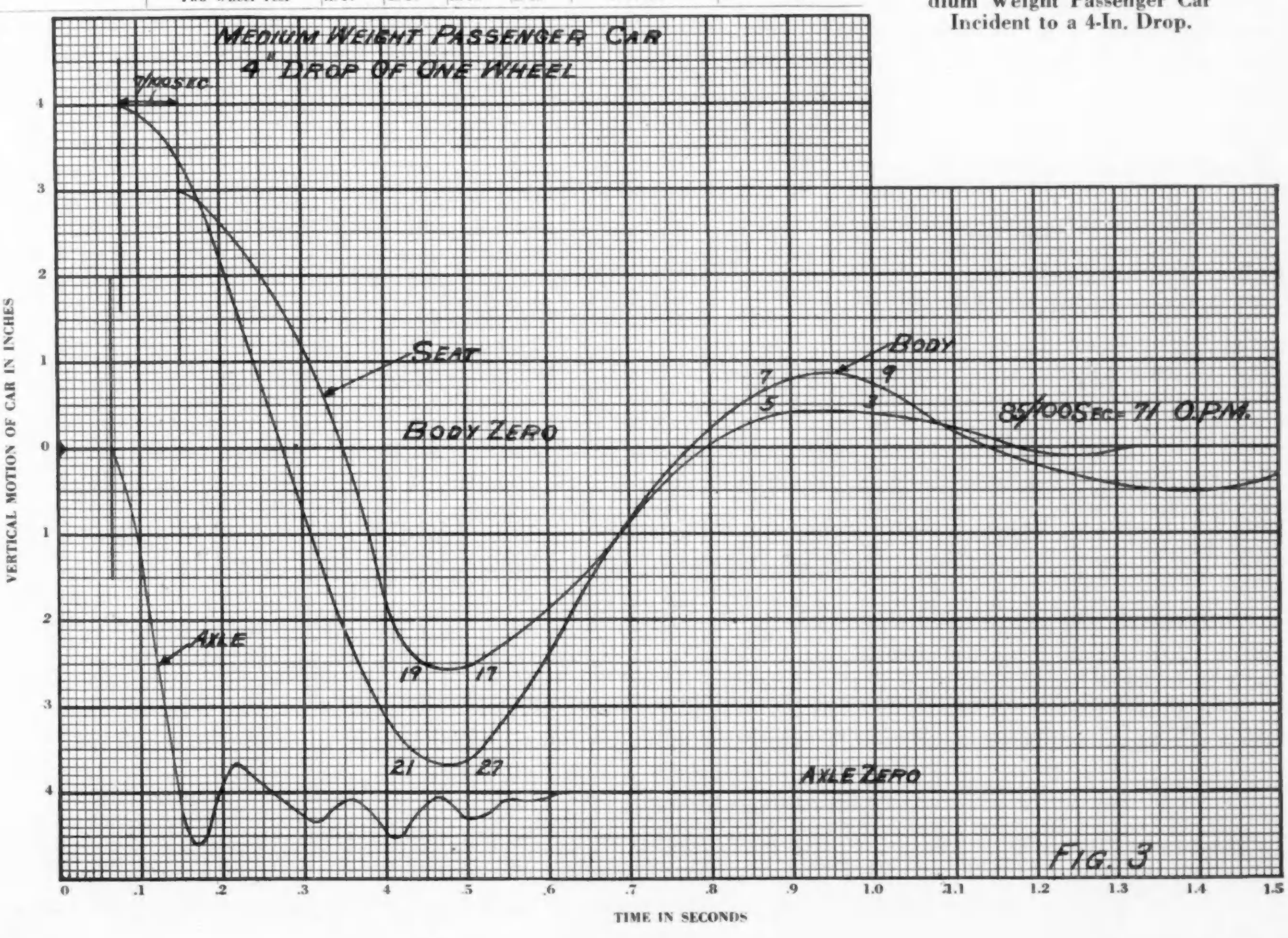


Fig. 3—Rear Wheel, Body and Seat Action of a Medium Weight Passenger Car Incident to a 4-In. Drop.

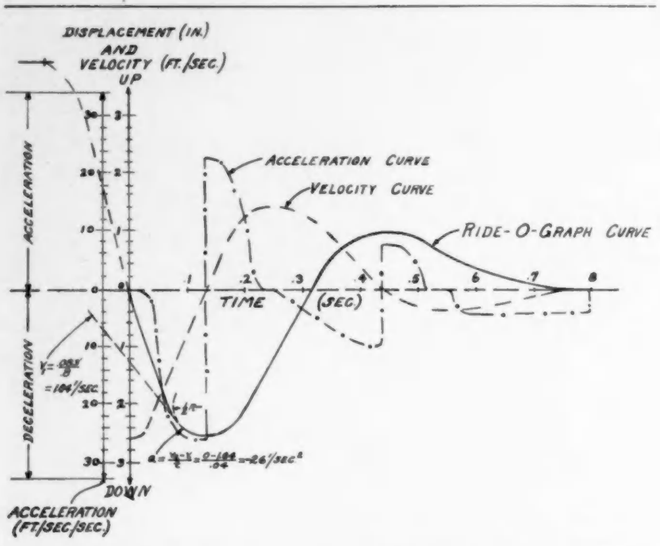


Fig. 5—Typical Body Curve with Velocity and Acceleration Curves Plotted by Graphical Differentiation

then be made for they are the result of the vertical motions. With very little experience, comparisons can be easily recognized. During the test the vehicle does not move along the floor but has only vertical motion. All cars are tested under identical conditions which is the first requisite of a reliable test. The impact of the wheel is produced by gravity, a constant. Hence, any difference shown in the graph is known to actually exist in the mechanism being tested.

If two given springs have a difference in their stiffness value, this relative difference will manifest itself under all conditions of impact. The same holds with tires. Since under a given set of conditions, each of these two elements produces a constant action individually, they will when working in combination produce a constant combined result. With shock absorbers, we do not have a constant, for many variables exist. The first depends upon the amount of work they are designed to do; the second, upon the adjustment (assuming the use of adjustable type); third, upon the time required for doing their work; fourth, at what point in the cycle the maximum resistance is applied. The effect of these variables can be materially reduced if the shock absorber and factory engineers give them sufficient attention and study. Ride-O-Graph charts disclose that a difference in the timing of the shock absorber of only 0.02 sec. can cause a great difference in the action of the body and the wheel.

Fig. 2 shows the accelerations in a typical medium-weight passenger-car body equipped with standard conventional leaf springs. The entire front end of the car was raised 4 in. and then dropped. Then the entire rear end was raised 4 in. and dropped. One curve shows the motion of the body immediately over the front axle incident to the drop of the front end. The other shows the motion of the body immediately over the rear axle incident to the drop of the rear end. The graphs prove that the highest accelerations occur as the motions accelerate from zero and decelerate to zero. If the accelerations and decelerations are reduced, the relative damage caused to each unit of the vehicle incident to them should be reduced correspondingly. The declaration figure of 560 ft. per sec. per sec. on the front curve is very high because of the low cushioning qualities of the tire, spring and shock absorber. The downward motion of the body was stopped too abruptly. The acceleration rate upward of 235 ft. per sec. per sec. while considerably lower than the deceleration, is still very high compared with the average conventional car. On the rebound motion above zero, the deceleration of 47 ft. per sec. per sec. is also high and in excess of gravity, which is 32 ft. per sec. per sec. This means that if the passenger were positioned where or directly above the point where this motion was recorded the passenger would leave the seat or the pay load would leave the body. The acceleration of 37 ft. per sec. per sec. downward is also faster than gravity and would pull the seat down faster than the passenger or pay load falls. On the rear curve, the decelerations and accelerations are very much lower mainly because of a longer spring having a vibration rate of about one half that of the front. High velocities are detrimental because they build up high forces which cause high amplitudes of motion. High amplitudes build high stresses in springs, tires and connecting units. High amplitudes almost invariably cause high decelerations and accelerations.

Professor Moss has been doing some very fine research pertaining to the physiological effect of these accelerations, velocities and amplitudes upon the human being. With the wobblemeter and other tests, he observes the effect on the passenger or driver, whereas in our research we have been endeavoring to determine the cause and extent of these discomforting and damaging motions. We have

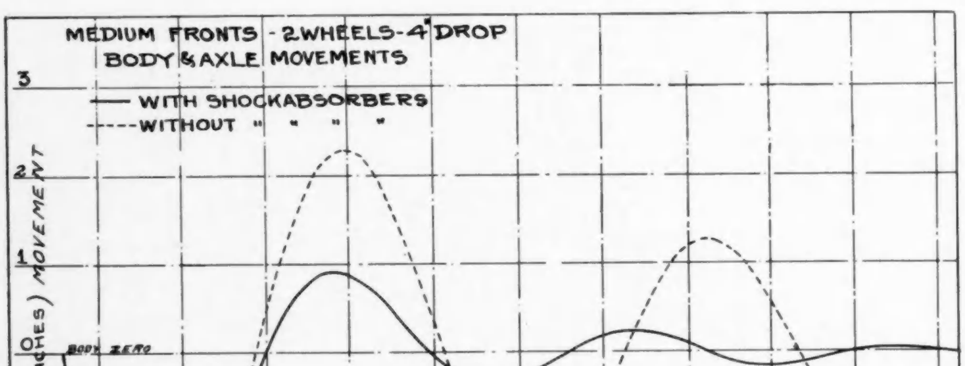


Fig. 6—Typical Body and Axle Movements

made uncounted changes in springs, tires and shock absorbers endeavoring to improve the performance of hundreds of vehicles from a comfort and maintenance standpoint.

Some qualified engineers state that any acceleration of the human body greater than 5 ft. per sec. per sec. is discomforting. Physiological tests made by Professor Moss, and tests we have made, all tend to prove this to be true. Although the industry since its inception has been talking about riding comfort, it now appears that no vehicle thus far produced really possesses comfort qualities because all give accelerations greater than 5 ft. per sec. per sec. In other words, they all produce discomfort to a greater or less degree. Anything which tends to reduce acceleration rate, velocities and amplitudes, tends to reduce discomfort. Vertical body motions that are discomforting originate at the tire and must be followed up through the springs, shackles, shock absorbers and the chassis to the seat cushion in passenger cars and to the pay load in commercial vehicles, if they are to be completely understood. In our research, we naturally give most attention to springs, but we also study shock absorbers, tires and shackles. Changes in seat cushions are not practical from a service station standpoint, but seat cushion tests on passenger cars have been made by the Ride-O-Graph and the information thus gained is definite and reliable.

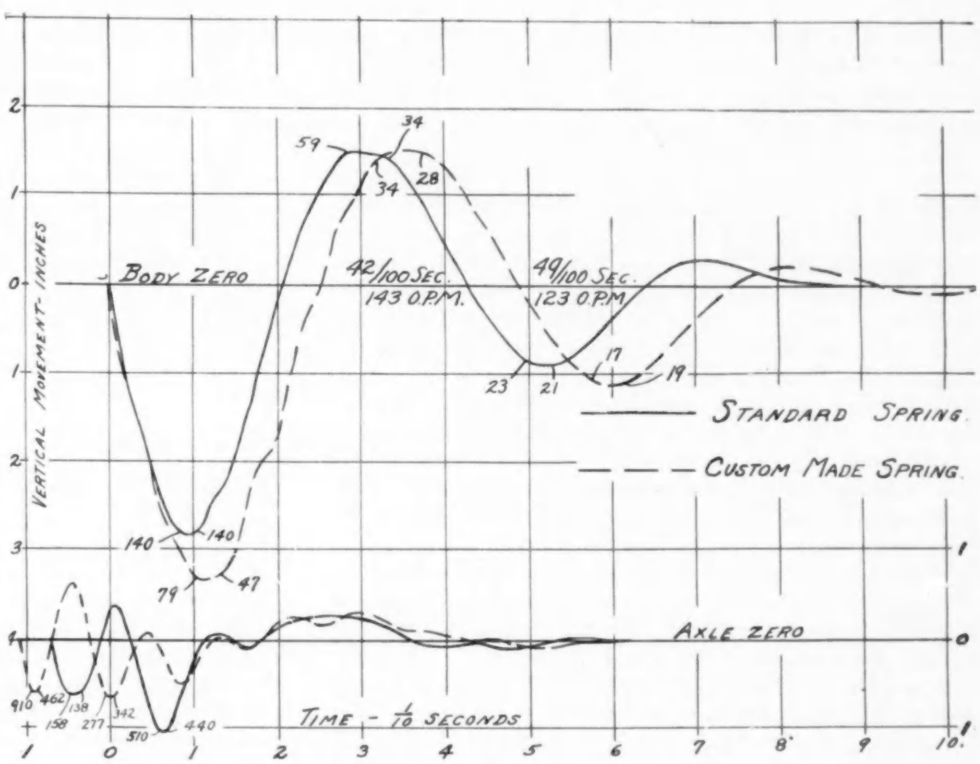


Fig. 7—Front, Body and Axle Movements of Popular Make 1½ Ton Truck with Hydraulic Shock Absorbers

Fig. 3 shows the rear wheel, body and seat action of a medium-weight passenger car incident to a 4 in. drop. While the body was lifted 4 in., the seat moved upward only 3 in. because it was nearer the axis, as the measurement of the lift in this test was made at the outside edge of the fender. The curves show that the deceleration rates as each reached maximum deflection were: for the body 21 and for the seat 19 ft. per sec. per sec. In the acceleration upward, the body shows 7 and the seat 5 ft. per sec. per sec. On the acceleration downward, the body shows 9 and the seat 3 ft. per sec. per sec.

Fig. 4 shows the fourteen important factors involved in the suspension problem, in so far as the motions of the wheels and body are concerned. Each one must be properly analyzed. The order or number of the respective items has no specific relation to their relative importance. No. 5 factor, the periodicity, particularly of the body, is of prime importance, but low periods in themselves do not by any means always produce best results as some have erroneously contended. Nos. 8 and 10 and subsequent deflections are of vital importance. These compression zones on the axle curves represent tire deflection and it is here where all suspension movements originate and where the discomforting, damaging and unsafe motions have their inception. We should give these tire motions our first attention so as to ascertain and become familiar with the forces with which we have to contend. Observation of these forces represents the foundation upon which we must base our analysis of the suspension problem. We should then follow these forces and the reactions they produce, up through the springs, shackles, shock absorbers, chassis, and on to the seat. It is not necessary to explain at this point the importance of each of these fourteen factors.

Fig. 5 shows a Ride-O-Graph space-time curve and two other developed curves, one a velocity curve showing the velocity of the body at any corresponding instant on the space-time curve. The instants chosen are indicated at points along the horizontal line. The other is an acceleration curve showing the acceleration of the body at any corresponding instant on the space-time curve. Both developed curves are derived by graphical differentiation. The space-time curve is graphically differentiated so as to secure points for plotting the velocity curve. The velocity curve is then graphically differentiated so as to secure points for plotting the acceleration curve. Many engineers are familiar with this process. Considerable time was spent in devising methods and means which would enable us to read these curves quickly, particularly with reference to the acceleration value. Upon going over several methods that were in use, we found that none were so satisfactory for practical purposes as the one we have adopted. Others had not taken into consideration certain factors which we found it was necessary to consider.

The Ride-O-Graph's curves of Fig. 6 show the motion of the body and axle with and without shock absorbers. Thus the effect of shock absorbers on body and axle is determined. The curves were taken on the front of a medium-weight passenger car. The effectiveness of the shock absorber is quite apparent, particularly in the decreased amplitude of body motion and the number of body oscillations. The shock absorber brought the body to rest much more quickly than did the spring by itself. A considerable difference also occurred in the axle movement with and without shock absorbers. The "danger line" was established after many tests were analyzed and road observations made in the matter of

braking and steering. It was found that any wheel with rebound above this danger line, that is, more than one-half inch above the zero line, created an unsafe condition, as braking and steering effectiveness depends upon the tires being firmly on the ground.

Fig. 7 represents a typical case in which the standard springs of a popular 1½-ton truck were replaced by custom-made springs. The oscillation rate on the custom springs was approximately 14 per cent lower than that of the standard springs. It will be noted, that the custom sprung axle reached the ground approximately 0.03 sec. sooner in relation to the body than did the axle with the standard springs. The deflection on the second cycle of the axle shows over 1 in. with standard springs, whereas with custom springs it shows but 0.6 in. This difference in deflection of body and axle is caused largely by the change in the body and axle cycle in their relation to each other. The nearer any axle motion downward approaches the body motion downward, as the body nears its maximum downward motion, the greater is the body and axle deflection. This is easy to understand for when the body and axle travel in the same direction, their combined forces are operative. The same situation applies on rebound motions upward. The opposite exists when the body reaches its maximum downward deflection at the same instant the axle reaches its maximum rebound movement. In this instance, the two forces oppose each other and one tends to counteract the other. In service work it is difficult to break up these objectionable synchronous tendencies of the body and axle. Many suspensions have these objectionable tendencies each time the wheel goes into a depression in the road. The vehicle manufacturers, by making certain changes in the suspension, should be able to show gratifying improvements in this particular.

	Deceleration			Acceleration		
	Standard Springs	Custom Springs	Per Cent Improvement	Standard Springs	Custom Springs	Per Cent Improvement
			First Compression			
Body	140	79	43	140	47	66
Axle	158	910	—470*	138	462	—230*
			First Rebound			
Body	59	34	42	34	28	18
			Second Compression			
Body	23	17	26	21	19	9
Axle	510	277	45	440	342	22

*Impairment.

Fig. 8—Comparison of Acceleration Figures Covering Fig. 7

The body figures under deceleration represent the relative cushion which the tires, springs and shock absorbers afforded to the front end of this body, incident to the impact, which in this test was a 4 in. drop of the entire front end. Standard equipment springs show 140 ft. per sec. per sec. deceleration; custom springs, 79 ft. per sec. per sec. Thus, the custom springs afforded 43 per cent greater cushion to the body incident to this impact, than did the standard springs. De-

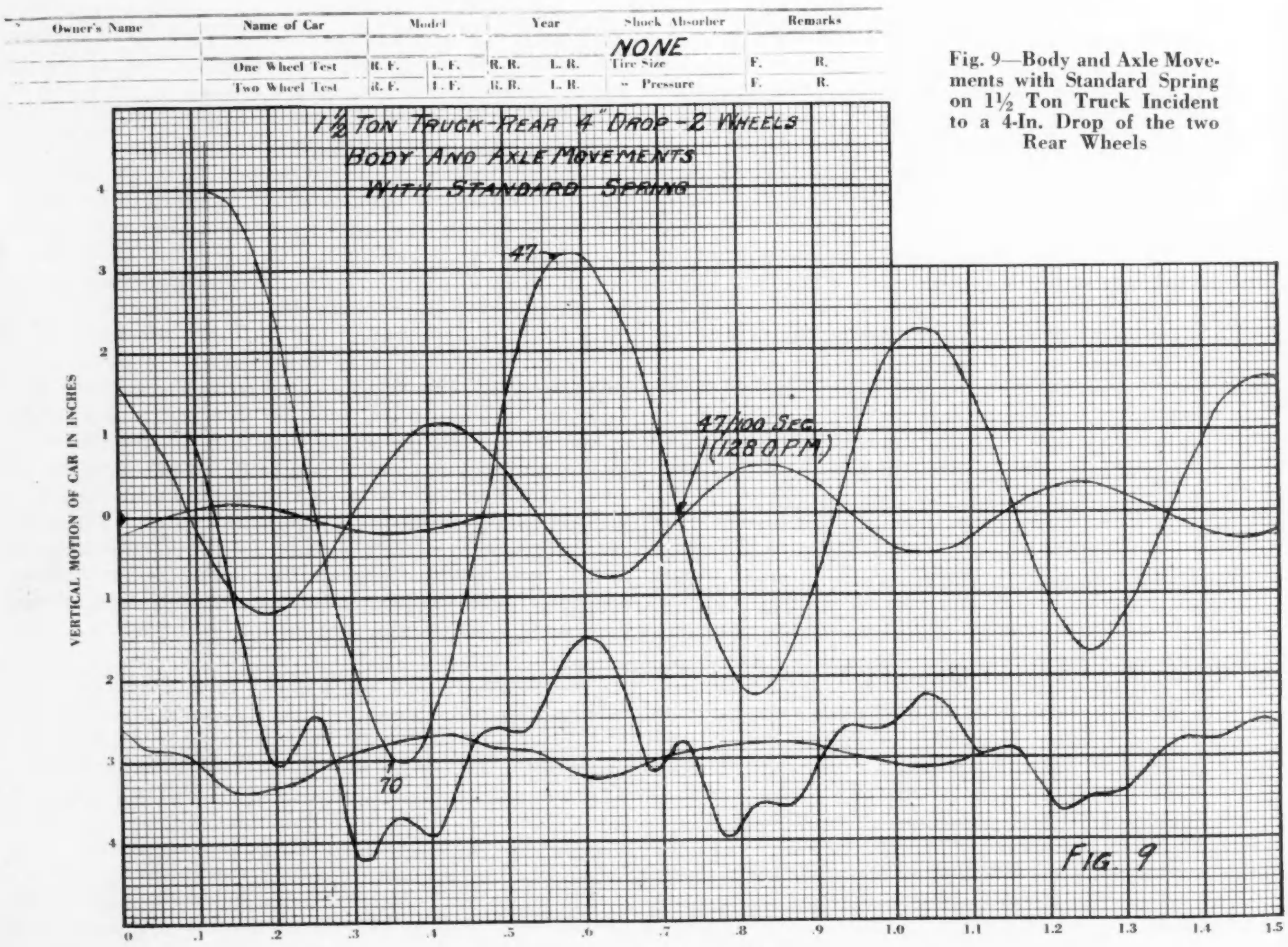


Fig. 9—Body and Axle Movements with Standard Spring on 1½ Ton Truck Incident to a 4-In. Drop of the two Rear Wheels

celeration rates on the fronts of different vehicles tested vary from 15 to 500 ft. per sec. per sec. On the body acceleration rate upward, after the motion downward has decelerated to zero velocity, standard springs show 140 ft. per sec. per sec., and custom springs 47 or an improvement of 66 per cent resulting from custom springs. Accelerations on these upward motions in tests made on uncounted cars, vary from 15 to 500 ft. per sec. per sec.

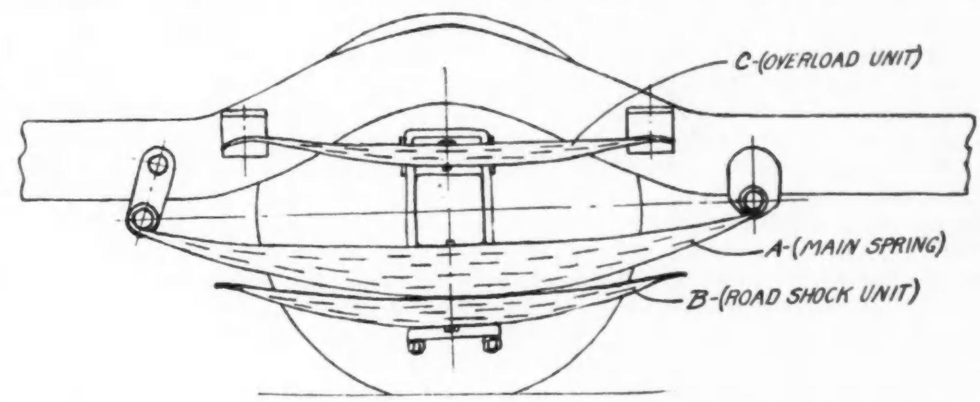


Fig. 10—Rear Spring Suspension (Underslung) Custom Built Cradle Compensating Main and Cradle Helper Combination on Bus Chassis of 12-25 Passenger Capacity

On the axle deceleration incident to the same test, standard springs show 158 ft. per sec. per sec. and custom springs 910. This inconsequential impairment by the custom springs was the result of their greater flexibility. Being more flexible, the custom springs permitted the tire to reach the ground sooner than did standard springs. This means that the custom sprung tire had less body weight on it at the instant it contacted the ground and this caused it to bounce more freely than did the tire with standard springs. These decelerations in different axle tests vary from 100 to 5,000 ft. per sec. per sec. On the axle acceleration motions upward, standard springs show 138 ft. per sec. per sec. and custom 462, an unimportant impairment by the custom springs. As might be expected, the wheel with the custom spring showed a higher rebound amplitude than did the standard spring, because of its high acceleration upward. These accelerations on the upward axle motion on different cars vary from 100 to 5,000 ft. per sec. per sec. Strange as it may seem, one of the heavy and highest priced foreign cars produced one of the highest axle acceleration motions thus far found. This very undesirable condition obviously produced discomfort.

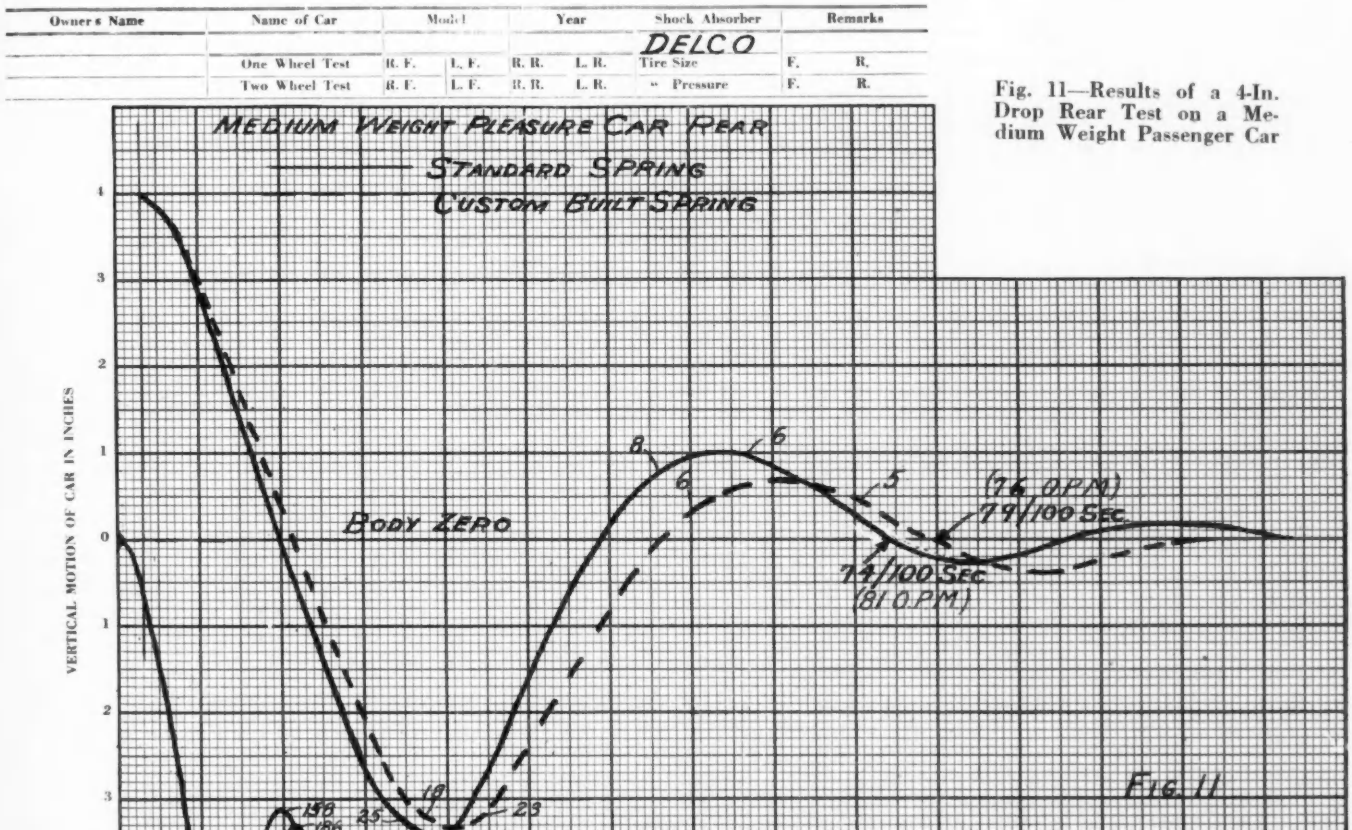


Fig. 11—Results of a 4-In. Drop Rear Test on a Medium Weight Passenger Car

Referring to Fig. 8, on the first rebound body deceleration, the standard spring shows 59 ft. per sec. per sec. and the custom spring 34, or an improvement of 42 per cent by custom spring. These figures vary in different cars from 3 to 100 ft. per sec. per sec. On the acceleration downward, the standard spring shows 34 ft. per sec. per sec. and the custom spring 28, or an improvement of 18 per cent by custom. These figures vary on different cars from 3 to 100 ft. per sec. per sec. Body rebound deceleration and acceleration motions are very important, for if either is faster than gravity, the passenger will leave the seat or the pay load will leave the body, assuming either is directly over such motion.

Thus far, we have not given any particular study to the deceleration and acceleration of the axle rebound motions. We realize, however, that the amplitude of the axle rebound is of great importance because braking and steering effectiveness are dependent upon the tire being in contact with the road.

The first oscillation in these body and axle curves produced by the Ride-O-Graph simulates the motion resulting from the wheel encountering a depression or hole in the road.

Depressions and holes rather than high spots in the road cause the highest accelerations to the body as the vehicle passes over them. The second and subsequent oscillations of the body as shown in these graphs, are important since they simulate the action that takes place when the tire encounters raised irregularities in the road. More is yet to be learned regarding all of these motions, particularly those pertaining to the axle. Just how much axle accelerations affect tire wear, we have not yet determined. It is certain, however, that high axle accelerations mean high external and internal friction of the tire. Certainly high deflections cause high stresses in the tire wall. Maximum effectiveness of the suspension can not be realized if the wheel is not permitted free movement downward, incident to slight depressions or holes in the road. More study should be given to the wheel and body motion as a working unit.

In view of these facts regarding the motions of the body and axle, should we continue to strive for a harmonic motion in the body as has been our inclination in the past? We now see there can be no true harmonic motion of the body in present suspension. Moreover, we also conclude that discord or heterodyning effects are desirable if best results are to be realized. Our problem in this particular, is somewhat the same as confronts the electrical, acoustical and radio engineers. Some time ago they realized heterodyning effects in certain instances were desirable and steered their course accordingly. Simple harmonic motion is defined, as the projection of uniform circular motion upon a diameter. To execute such motion, the force of resistance must be proportional to the displacement and oppositely directed. Therefore we must realize, as did the engineers in other fields, that our problem in automobile suspension is much more complex than simple harmonic motion. Harmonic motion is the ideal, a goal for which we may strive but which we will never reach. So with comfort. It is a goal, an ideal. Perfection is unobtainable, hence we should establish a reliable "yardstick" to tell us definitely which way we were headed, whether toward or

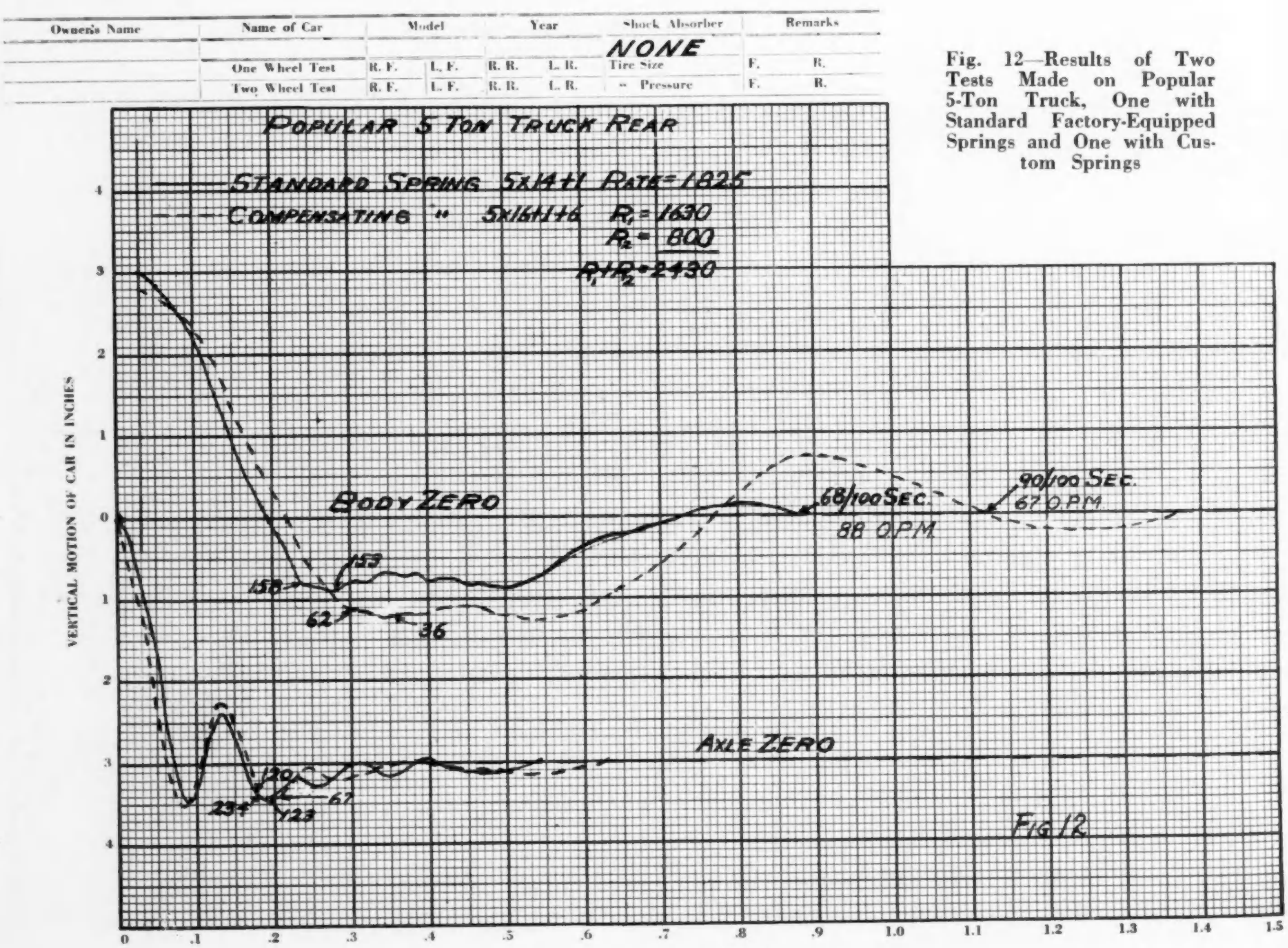


Fig. 12—Results of Two Tests Made on Popular 5-Ton Truck, One with Standard Factory-Equipped Springs and One with Custom Springs

away from our goal. Discomfort should be established as a known measurable quantity and we should strive to reduce it quantitatively in the suspension. A change of viewpoint or objective, pertaining to a given problem, frequently clears the path and accelerates progress. If this paper causes a change in viewpoint regarding suspension problems and results in some progress toward their solution, the author will consider his efforts have been worth while.

The Plane of Travel

The greater the number and the greater the amplitude of the vertical movements the vehicle makes in its course of travel from one point to another, the greater is the distance it actually covers. It is roughly estimated that the average vehicle when traveling over the average road, covers at least 10 per cent greater distance than if it were not subjected to vertical motions. Each time a vehicle and its load moves up against gravity, a certain amount of energy is required. The greater the amplitude, the greater the energy required. This means greater fuel consumption. Thus far we have not secured any satisfactory quantitative data in this regard, but measurements already made of the energy required to propel a given vehicle a given distance over a smooth road and over a rough road, indicate a difference of from 5 to 17 per cent in favor of the smooth road. Therefore, if a vehicle and its load can be made to travel in a more nearly even plane, similar savings in energy expenditure should be realized.

High amplitudes of body movement tend toward unsafe operation conditions. Tests show that high rebound amplitudes of the body tend to lift the tires off the ground and this of course, reduces braking and steering efficiency. When the body amplitudes downward are high, they cause high deflections in the spring and tire which mean high spring and tire stresses, which frequently cause spring and tire failures. Springs that are too stiff for a given suspension, tend to lift the wheels from the ground thereby impairing the safety of the vehicle.

The curves in Fig. 9 show the body and action motion of a popular $1\frac{1}{2}$-ton truck incident to a 4-in. drop test on the rear. The first rebound amplitude is greater than the first compression amplitude. This is very unusual and indicates a very unsafe condition. The axle curve shows that the wheel was lifted off the ground almost 1 in. Seven complete oscillations of the body occurred before it came to rest. They resulted in great discomfort to the driver, much damage to the vehicle and pay load, and also in unsafe roadability. The main cause of this undesirable condition was that the front springs were too stiff for this job and the rear helper springs contacted at body load. Moreover, as is seen by these curves, the body and axle are synchronous in their upward motion which produced high velocity which, in turn, resulted in high rebound amplitude.

Standard Method of Testing

After two years spent in making tests on various cars, taxicabs, trucks and buses, we were able, about a year ago, to establish definite, standard methods of testing and to arrive at allowable or desirable maxima in amplitudes, velocities

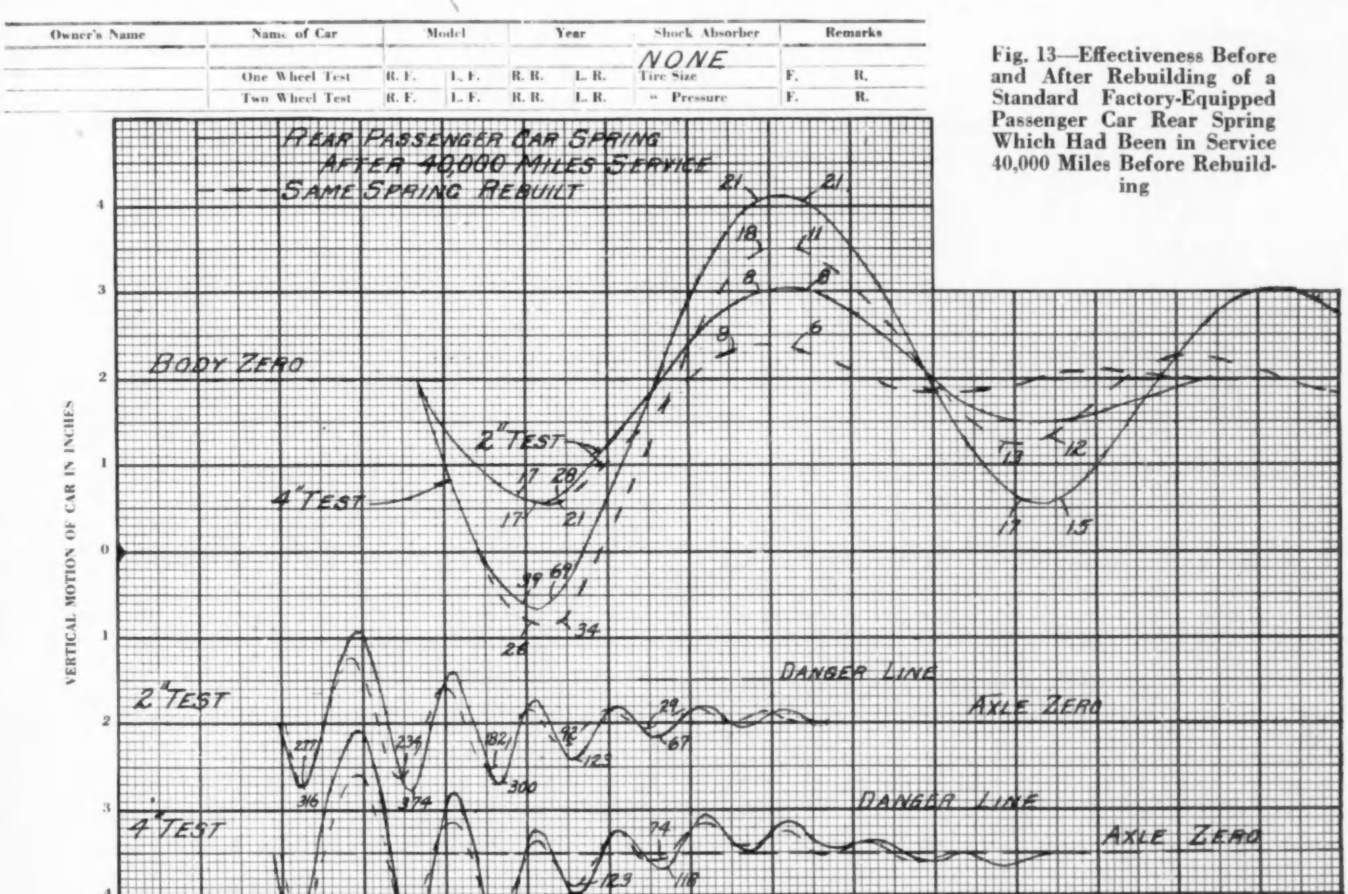


Fig. 13—Effectiveness Before and After Rebuilding of a Standard Factory-Equipped Passenger Car Rear Spring Which Had Been in Service 40,000 Miles Before Rebuilding

and accelerations of the body and all spring suspended parts above the axle and also for the same factors in respect to the axle, wheel, tire and other unsprung parts, at each of the four corners of the vehicle. We now definitely determine the forces set up in each tire incident to any road shock of any intensity, operating under any given sprung and unsprung load. We find that tires vary greatly in their dynamic action, in amplitude, velocity and acceleration. Tires are studied first, for once we know their reaction to various road inequalities, under a given sprung and unsprung load, we are then able to proceed intelligently with the spring and the shock absorber problem. Hence, first the tire action is determined, then in succession that of the spring, the shock absorber and the entire combination. By this procedure, we approach the combined problem in its simplest form and eliminate guesses and opinions. During the process, we determine the true relationship of the body cycle with the wheel cycle, which is of great importance.

We learn what part the tires, springs and shock absorbers play in respect to comfort, also what effect each has on braking and steering. We hope in the near future, to predict reliably the relative maintenance cost in the matter of tire wear, also the general wear in the various chassis units, and the safety of the wheel action as affected by the suspension as a whole. Some of the changes we have thus far made, have resulted in very gratifying improvements. In some cases tire wear has been increased 100 per cent. This, of course, is exceptional and not to be regularly anticipated, by any means. On one taxicab fleet a phenomenal reduction of 32 per cent in general maintenance was shown as a result of a change made in the springs. In other instances, the safety of the vehicle was materially increased and brake troubles were solved.

Fig. 10 shows a compensating rear spring combination for heavy commercial vehicles, particularly those carrying a widely varying load. Segment "A", is the main unit similar to the conventional type spring. "B", is a supplementing unit which operates in conjunction with the main spring. "C" is the overload unit. We have, with this combination, been able to reduce materially body and axle accelerations and to secure a much safer wheel action than obtained with the conventional standard spring, having but one main unit or with a main and helper. The segment "B" takes the larger road shocks and since it has a rolling contact with the main unit, it operates only to the extent required. The greatest advantage derived from this compensating combination, is realized when the vehicle is empty or carrying a light load. On some standard factory equipped jobs we were able to cut the stiffness rate of the conventional main spring almost in half. Few operators realize that greatest damage to the vehicle, occurs when it travels empty or with a light load. Accidents, according to insurance records, also occur more often when the vehicle is empty than when it is fully loaded. In the average commercial vehicle, the miles covered when empty or light, are greater than when fully loaded. In taxicabs, 72 per cent of the mileage is when empty and less than 0.1 per cent is with a capacity load. It hardly seems conceivable that engineers will continue to install springs of the present standard type, or operators continue to operate them, when the loads

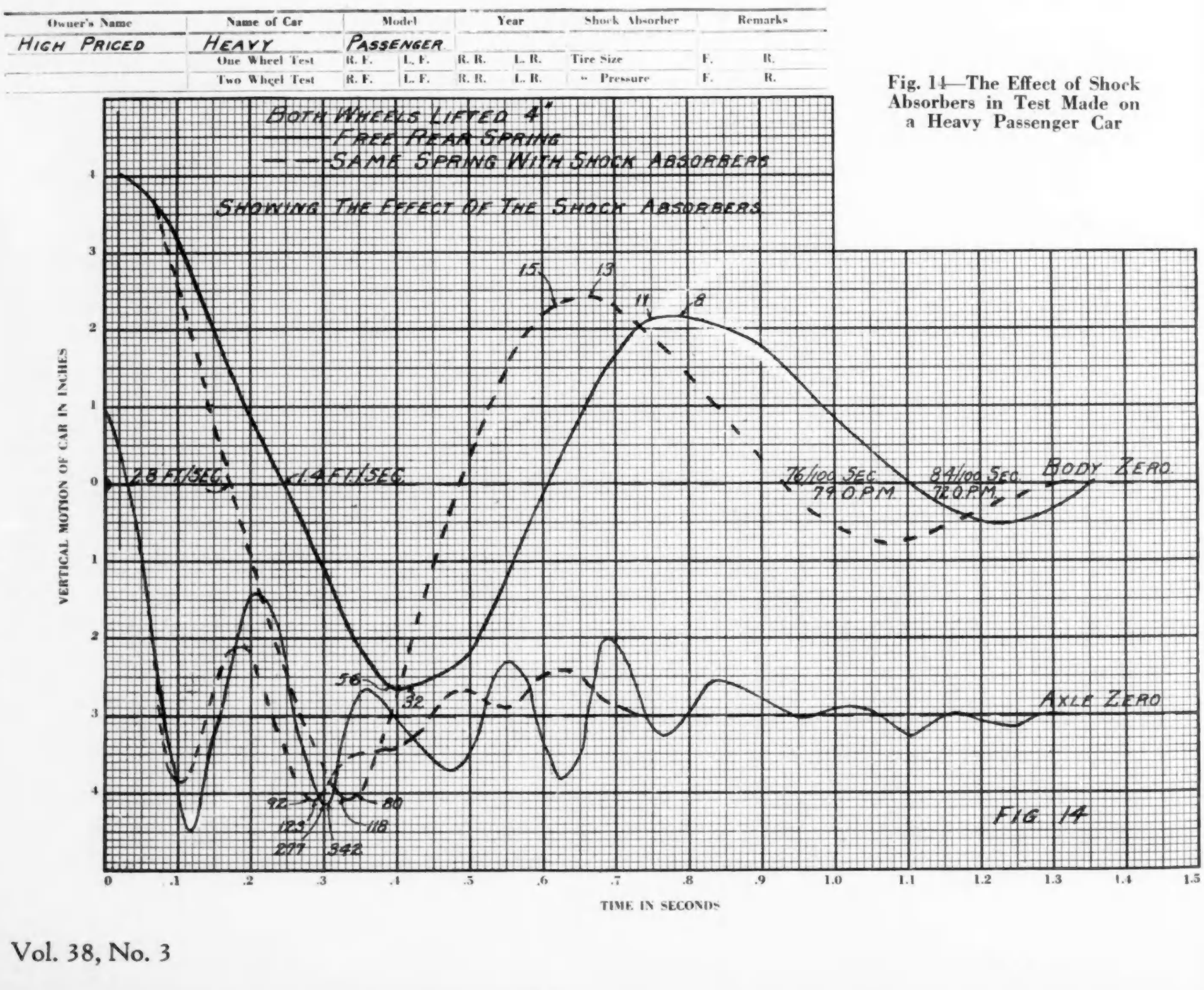


Fig. 14—The Effect of Shock Absorbers in Test Made on a Heavy Passenger Car

vary widely, once they realize the distinct advantages which this compensating type of spring suspension affords in comfort, safety and economical operation.

Fig. 11 shows a 4 in. drop rear test on a medium-weight passenger car. One curve shows the standard factory spring, the other shows a custom built compensating spring. The body oscillation rate with the standard spring shows 81 and with custom, 76 oscillations per min. First body deceleration with the standard spring shows 25, and with the custom 18 ft. per sec. per sec. which means that the custom spring afforded 28 per cent greater cushion. On the acceleration upward, which effects discomfort and damage to vehicle, the standard spring shows 35 and the custom spring 23 ft. per sec. per sec., which means 34 per cent improvement with the custom spring. On body rebound, decelerations and accelerations, the custom spring shows a 21 per cent average reduction. The axle amplitudes were slightly reduced by the custom spring and the axle accelerations were reduced an average of 12 per cent.

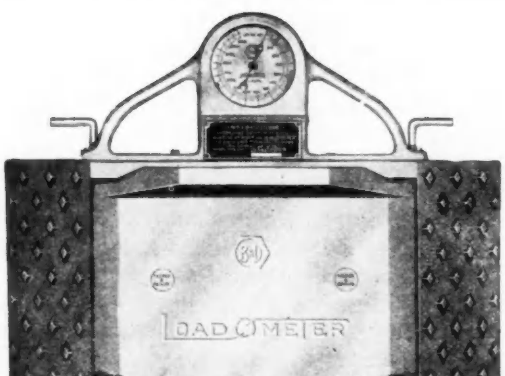


Fig. 15—Standard Road Scale-Lodometer

The curves in Fig. 12 show two tests made on a popular 5-ton truck. One with standard factory equipped springs and one with custom compensating springs. The test was with a 3 in. drop on one rear wheel. In lifting the wheel 3 in. incident to the test, the body with standard springs raised 0.2 in. higher than with custom springs. The body shows a periodicity with the standard spring of 88 and with the custom spring, 67 oscillations per minute, an improvement of 24 per cent for the custom spring. Body amplitudes were considerably increased by the custom spring, however, contrary to what usually happens, the average acceleration was reduced 68 per cent, which is a tremendous improvement for the custom spring. The average axle accelerations on the second compression were reduced 48 per cent by the custom spring. These distinct improvements were accomplished despite the fact that the custom spring provided greater carrying capacity yet its maximum stresses were no higher than they were on the standard spring.

In about six months, we shall have actual performance records covering vehicles on which we have made various changes and we have every confidence that the maintenance costs on these vehicles will be substantially reduced.

We have found marked differences in the dynamic action of leaf springs though they statically tested the same in their stiffness value and had the same number of leaves of the same length and thickness. In other words, springs of the same specifications as to physical dimensions, we now know may vary considerably in their dynamic values. We also

ENGINEERING REPORT NO. Date——————

To
Customer
Vehicle
Complaint: By—

Carbon Copies to
Party Contacted

INSPECTION: STANDARD FOUR WHEEL VEHICLE ( ) TRACTOR ( ) TRAILER ( ). Weighed and inspected the above vehicle, Empty ( ), Loaded ( ). Computed F ( ) R ( ) Helper ( ) Spring Steel Stresses and from these figures determined the Safety Factor against breakage on each spring. Ascertained the body oscillation periods and amplitudes, also computed the accelerations, all three being direct factors concerning the safety and comfort of the vehicle. The extent of the amplitudes and accelerations is also a direct barometer of the "wear and tear" resulting therefrom, to the tires, axles, chassis, body and payload. The extent of these amplitudes and accelerations is also a direct barometer covering the longevity or depreciation of the vehicle as a whole.

FINDINGS:

Stresses in the Spring Steel:

Front	M	L	H
Rear	M	L	H
Helper	M	L	H

Oscillation Frequency of Body:

Front	M	L	H
Rear	M	L	H

Amplitude of Body Movement

	Compression			Rebound		
Front	L	M	H	L	M	H
Rear	L	M	H	L	M	H

Amplitude of Wheel Movement

Compression			Rebound		
L	M	H	L	M	H
L	M	H	L	M	H

Accelerations of Body

Front	L	M	H
Rear	L	M	H

Accelerations of Wheel

L	M	H
L	M	H

Safety of Wheel Action

Front	H	M	L
Rear	H	M	L

Damage to Vehicle

Front	L	M	H
Rear	L	M	H

Discomfort Factor

Front	L	M	H
Rear	L	M	H

(Note)—"H" means high; "L" means low; "M" means medium. "N/T" means, no test, "C" means computed. The first letter in each case indicates the most desirable condition.

This report must not be construed in any sense to be a criticism or condemnation of the manufacturer who made this vehicle. It must be thoroughly understood that the manufacturer must make a vehicle to meet the average operator's conditions. He must establish and adhere to certain standards in production in order to deliver a standardized product on a most economical basis. It is conceded that operating conditions probably vary more with the springs, than with any other important part of the vehicle. Moreover, it must be borne in mind that springs through use get out of adjustment and need service the same as any other moving part of the vehicle.

Fig. 16—Engineer's Report Covering Suspension Test

know that the dynamic action of a given spring while in use on the car may vary considerably because of a variation in the condition of the bearing surfaces of the leaves, depending upon whether they are dry, rusty, smooth or rough, the manner in which the leaves were originally fitted, and the tension which existed in each leaf at the time the spring was originally bolted together. Some springs change more than others after a given period of service. As with other moving mechanisms, they show the result of wear. Some leaves get out of adjustment in their relation to each other. Other leaves lose their original tension. Any such changes in the spring naturally change its dynamic values.

Repairing and Rebuilding of Springs

Years ago, tire and shock absorber manufacturers acknowledged that their products could be repaired and rebuilt to the economic advantage of the owner. Some spring manufacturers still maintain, however, that springs differ from any other part of the car and that it is not economical to repair or rebuild them. Maintenance men, however, know from experience that it can be done economically and successfully. The problem of spring repairing and rebuilding is an important one. When a spring breaks, the first question to be answered is, "Shall a new spring be installed or the broken one repaired?" If the spring is broken in half, a repair is out of the question. On the other hand, if one half of the steel is not broken, contains no serious cracks or is not damaged in any other particular, there is no economic reason why such steel should be scrapped. In putting this used steel back into a spring, however, if each process in the procedure is not safeguarded, a satisfactory outcome is not assured.

The curves in Fig. 13 show the effectiveness before and after rebuilding, of a standard factory equipped passenger car rear spring, which had been in service 40,000 miles before rebuilding. The difference in body and axle amplitude and acceleration is very apparent. On the 4 in. drop test, the body deflection amplitude on the first compression is 8 per cent greater after rebuilding. The first rebound amplitude shows 29 per cent less on the rebuilt spring. The second deflection shows 52 per cent less amplitude on the rebuilt. On the 2 in. test, the first deflection amplitude is substantially the same. On the first rebound, the rebuilt spring shows

60 per cent less and on second deflections, the rebuilt spring, shows 70 per cent less.

On the axle amplitudes on the 4 in. test, the first deflection of the rebuilt spring shows 20 per cent less than before rebuilding. The first rebound, rebuilt spring shows 36 per cent less. On the second deflection, the rebuilt spring shows 28 per cent less and on subsequent amplitudes, the per cent of improvement by the rebuilt spring runs about the same.

On the body accelerations, the 4 in. test shows deceleration rates on the first deflection for the old spring 39 and for the rebuilt 26 ft. per sec. per sec. This means 33 per cent more cushion in the rebuilt spring. On acceleration rate upward on the first deflection, the old spring showed 69 and rebuilt 34 ft. per sec. per sec., an improvement of 50 per cent in favor of the rebuilt spring. The first rebound downward shows the acceleration rates were: old spring 21, rebuilt 11, or an improvement of 47 per cent as a result of rebuilding. On the second deflection, the rebuilt spring shows an improvement of 23 per cent in deceleration and 20 per cent in acceleration. On the 2 in. test, body accelerations are improved almost as much. On the axle accelerations, the improvements run from 12 to 57 per cent in favor of the rebuilt spring. Over all, the average improvement shown by the rebuilt spring in body accelerations was 31 per cent and in amplitudes 41 per cent. In the axle, accelerations were improved 32 per cent and amplitudes 29 per cent.

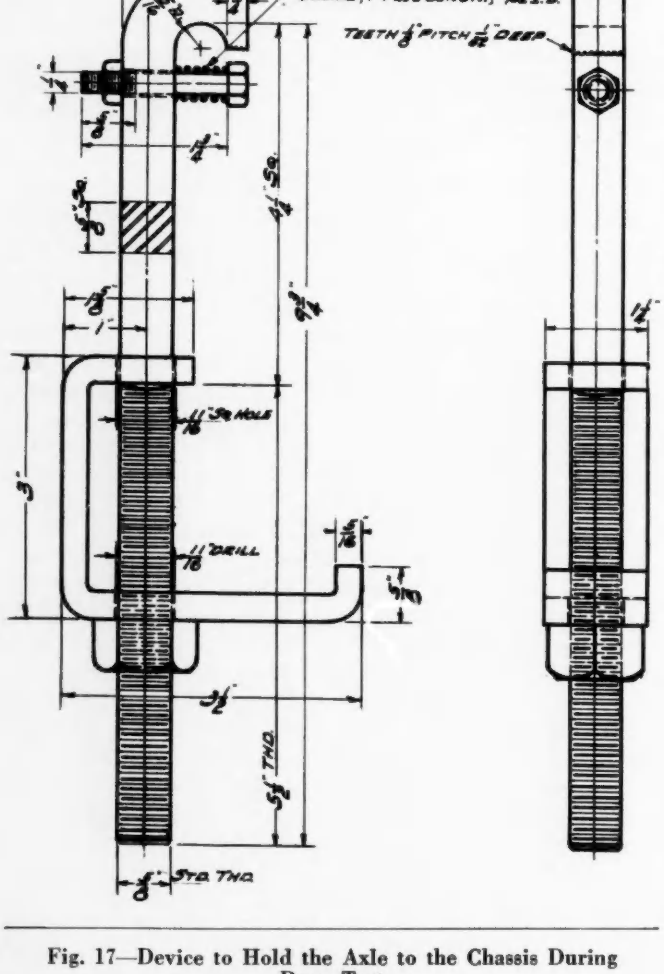


Fig. 17—Device to Hold the Axle to the Chassis During Drop Test

When operators realize that such reductions in amplitudes and accelerations directly reflect themselves in actual savings in vehicle maintenance, they will then give greater attention to the condition of the springs on their vehicles. At present, very few operators give the springs any thought until one breaks, even though most of them long ago realized that it pays to overhaul, rebuild or adjust other important units of the vehicle after a given period of service.

On the other hand thousands of leaf springs are being repaired and rebuilt and in consequence fleet owners are able to show worthwhile savings. Breakage when experienced with a repaired or rebuilt spring, is not always the result of faulty repairing or rebuilding. Many times it is because of faulty original design or because the spring was subjected to use not intended by its designer. Very frequently we find operators subjecting springs to use very far from the intentions of the designer. This applies particularly to overloading. On the other hand, it is not unusual to find springs in operation where the load upon them is materially less than the designer calculated them to carry. These misfits occur because the manufacturer must standardize his product to a greater or less degree, owing to economies involved in its manufacture and distribution. Many of these misfits could be avoided by the operator if he would take the trouble to ascertain what the proper springs were, for his particular set of conditions and then have such springs installed.

It has always been acknowledged by factory engineers that of two cars coming off the production line, one following the other, and both having the same springs, tires and shock absorbers, one often rides acceptably and the other not nearly so well. With the Ride-O-Graph tests, we readily locate the cause of these differences. Most of it is found to be in the springs and shock absorbers, yet some suppliers of these elements and some factory engineers are still reluctant to believe these facts. They still maintain that the free vibration period produced by a given spring on a given car is the true measure of the spring's effectiveness. Many refuse to acknowledge that forced vibrations are the major factor. As a matter of fact, while in actual operation on a car and working in conjunction with the tire and shock absorber, there is practically no such thing as a free vibration of the spring. An example of a free vibration is any pendulum movement. A spring's vibration period on the car is dependent upon its stiffness rate and the weight that is upon it. When the spring is operating dynamically on the car, it is compelled to operate under an entirely different condition than obtains with a free pendulum motion. The spring during its normal vibration period on a car, has to contend with intermittent forces set up by the tire. The extent and frequency of these forces depends upon the weight of the sprung and unsprung mass, also the acceleration, velocity and amplitude of the motion.

Since the period of vibration of the unsprung mass averages around five times the period of vibration of the sprung mass, there may be five forced motions of the spring, caused by the unsprung mass during one cycle of the sprung mass. Hence, it is perceived there are many more forced vibrations in the suspension than free vibrations. Moreover, there are times when the vehicle is traveling that the body is substantially at rest vertically but the wheels are moving. Such motions of the unsprung mass cause forced vibrations of the springs. It seems logical that improvements in the suspension will follow as the problem is made clearer.

The curves in Fig. 14 cover a heavy passenger car and show how badly the shock absorbers impaired the body motion. They increased the body velocity downward, compared with the free spring motion, from 1.4 to 2.8 ft. per sec. per sec.

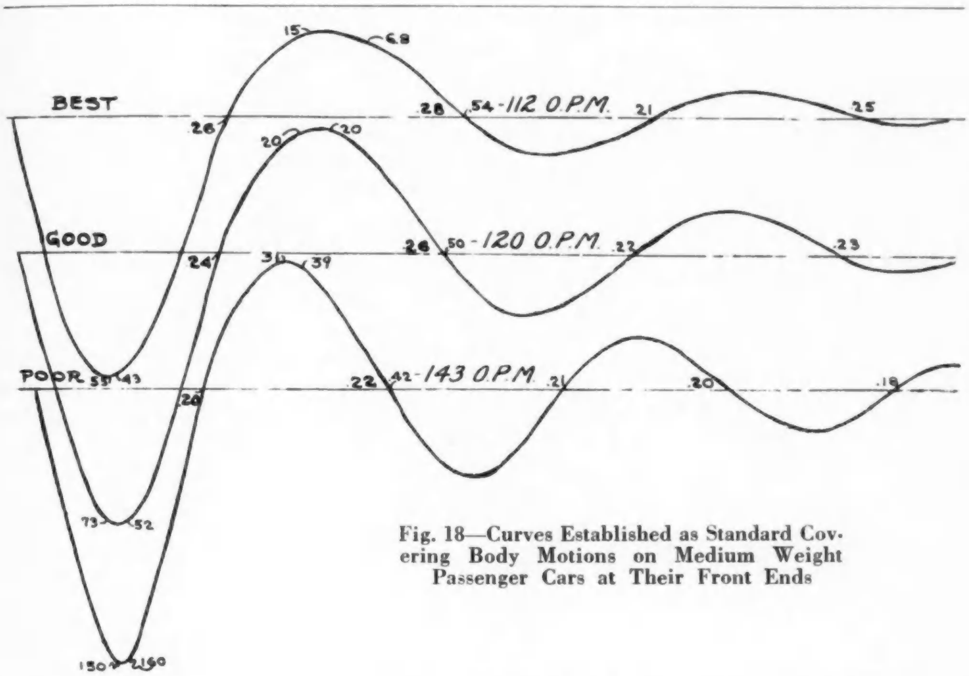


Fig. 18—Curves Established as Standard Covering Body Motions on Medium Weight Passenger Cars at Their Front Ends

Since kinetic energy equals ½ the mass times the velocity squared, the shock absorber increased the energy to be absorbed 300 per cent. In other words, the shock absorber had the opposite effect on the body from that which was expected of it. As a result, the deceleration, or cushioning effect, of the free spring was impaired over 100 per cent, the decelerating rate on free spring body motion being 56 and with shock absorber 118 ft. per sec. per sec. The increase in body velocity downward caused by the shock absorber, increased the body deflection 1.4 in. or 52 per cent. The shock absorber also increased the body oscillation rate from 72 to 79 oscillations per minute, or 10 per cent. It also increased the body rebound amplitude which was the result of high deflection, which in turn, created a higher acceleration upward. The shock absorber was unable to control this high acceleration, which it created and in consequence, the body rebound motion was higher than it was with the free spring. Many shock absorbers and rebound devices have similar impairment effects upon the free spring motion of the body. The axle motion in this case was materially improved by the shock absorber, but its ill effect on the body motion more than nullified the improvement it caused on the axle motion.

Too much emphasis and effort have been put on the theory that shock absorbers can perform the function of the springs. Unquestionably, the freer the spring is to perform its function as a cushioning element, the better will be the result. The name "shock absorber" as applied to many of the devices in use is a misnomer. On many cars, they perform no such function. Being single acting, they merely control the body rebound motion. If more cushion were to be put into the spring, which is easily possible at a slight additional cost, materially less work would be required of the control device. The spring has two definite functions. The first is to act as a cushioning element between the sprung and unsprung mass. The second is to keep the tires in contact with the road. The spring's effectiveness in affording cushion is directly measured by its ability to reduce the accelerations of the sprung and unsprung mass, and to control the amplitude of motion of each. The Ride-O-Graph shows clearly the spring's effectiveness in these particulars. When a road impact occurs, the degree of cushioning that the cushioning element affords depends upon the time consumed in the process. The greater the time consumed in dissipating the energies involved, the greater is the cushioning effect. Another element, namely, the distance through which the cushion is deflected, is also a factor. The main factor, however, which determines the cushioning effect of springs is time, because, in present automobile construction, the distance a spring can travel is limited.

Most of the shock absorbers and rebound devices on today's vehicles are badly overworked and in consequence, demand a great deal of attention. Many frequently have to be replaced with new units. With coil springs, these devices have to do almost twice the work they do when working in conjunction with the conventional leaf springs. That these devices have

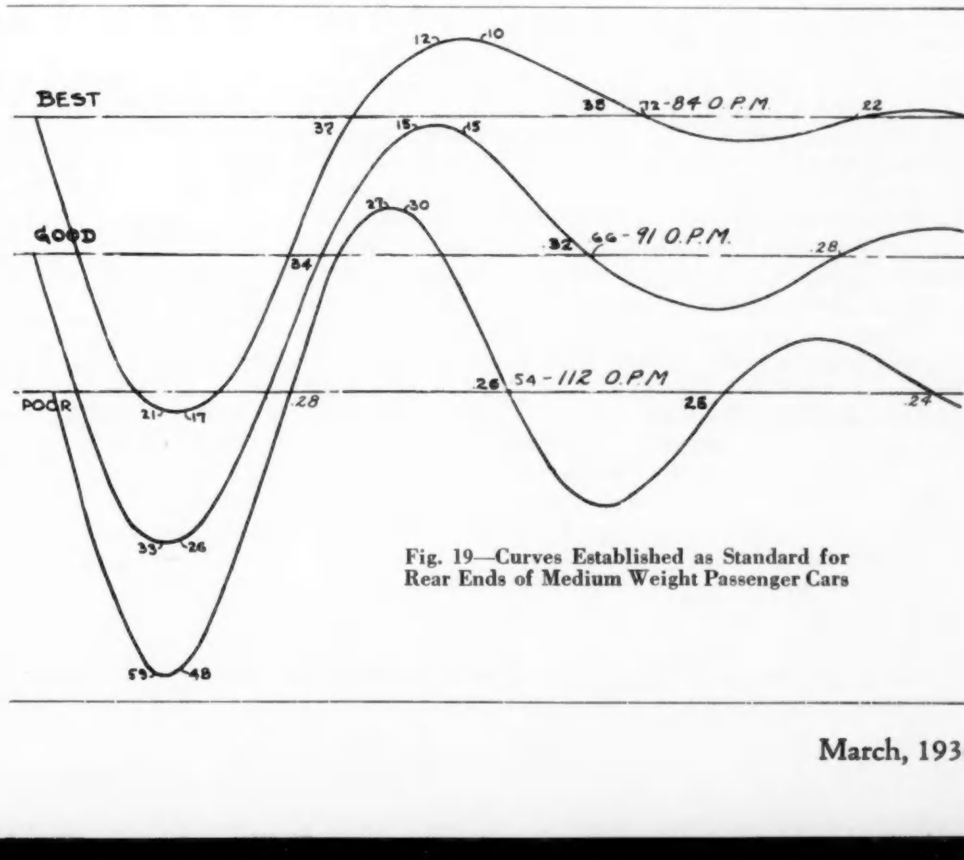


Fig. 19—Curves Established as Standard for Rear Ends of Medium Weight Passenger Cars

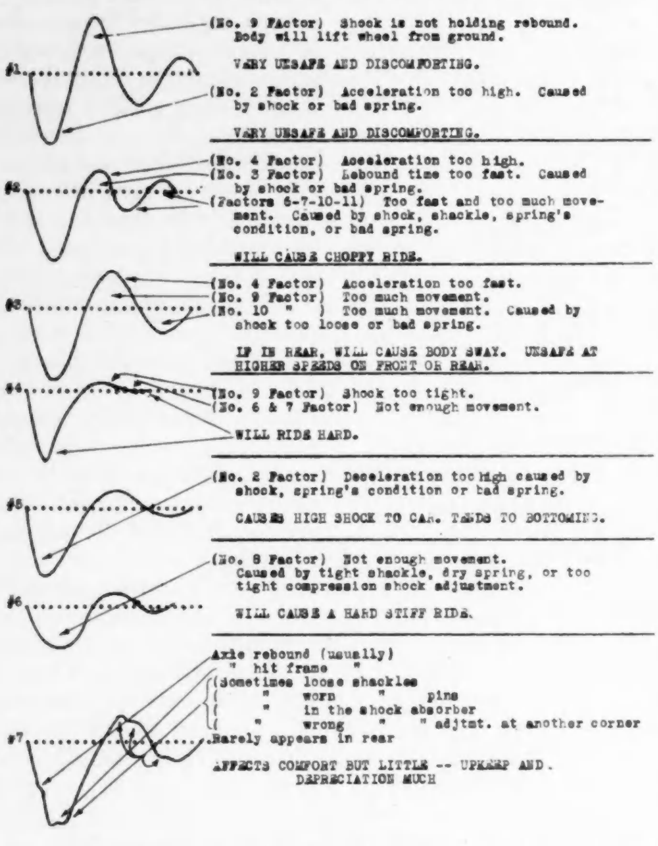


Fig. 20—Typical Complaints with Their Causes

stood up as well as they have, is a tribute to their manufacturers. If more cushion were put into the spring, it would produce less rebound force and thereby require less work on the rebound control device. Many owners, particularly commercial car owners, are complaining about these hydraulic devices because the expense of their maintenance is so high. Some of the taxicab fleet owners have become discouraged and removed all shock absorbers. Vehicle owners deserve something better. It can and should be given to them.

Springs Can Be Improved

Design: This factor is of paramount importance. Even a spring containing steel of the finest physical qualities, if it is not properly designed so as to meet the conditions under which it operates, will prove unsatisfactory. In leaf springs, this factor presents the greatest avenue for future improvement in cushioning as well as in endurance.

Type of Steel: Where maximum strength is desired, alloy steel of course will be used but it is misleading to infer that the analysis of the steel is the main factor governing a spring's ability to endure. It is known by all metallurgists that heat treatment is of greater importance. Tests made in England years ago by the Department of Scientific and Industrial Research, proved conclusively that the surface of a piece of spring steel whether for a coil or leaf spring, is of greater importance as effecting the strength and endurance of a given spring, than is the analysis of the steel itself.

Heat Treatment: This is a very important factor since it directly influences the structure of the steel itself and gives it the various endurance properties. The automotive industry since its inception, has shown constant progress in the development of alloyed steels as well as in their heat treatment.

Endurance Qualities: Much is yet to be learned with reference to the different characteristics that influence directly or indirectly this important factor. Heretofore, with reference to the steel, it has been a question of hardness, elastic limit, yield point, elongation and reduction of area. Fatigue and hysteresis have entered into the picture but much is yet to be learned regarding them. The author has devoted considerable time in developing tests on various materials and years ago, discovered two characteristics or properties hitherto unrecognized, that are common to all matter. Both are very important in any material where wearing qualities or endurance is a factor. They were named "recoverance" and "molinertia". To date little information of commercial value regarding them is available for the reason that no instrument thus far devised satisfactorily measures them. Where there is work done on a material, either by compression, tension or bending, it is important to know just what becomes of this work. There are three properties involved; recoverance, molinertia, and hysteresis, and the sum of these must equal the work done. Recoverance represents that portion of the energy which the material gives back immediately. Molinertia is that portion which the material gives back ultimately and hysteresis is that portion which is dissipated, presumably by molecular friction, and is never returned. Acceleration is a big factor affecting spring endurance but much is yet to be learned in this regard. The author is now working on a thesis titled "where the rate of recoverance equals or exceeds the rate of acceleration, there will be no fatigue". The design of the spring has much to do towards endurance, but it has had little attention. The effect of present day high powered brakes, is a big factor affecting spring endurance but has yet to be evaluated.

Construction

In the manufacture of a leaf spring there are six main factors that influence its cushioning and endurance qualities. (1) condition of all working surfaces, (2) condition of the bearing surfaces, (3) tension between each leaf, (4) shape of the ends, whether tapered or full thickness, (5) number of leaves, (6) thickness of the leaves. A material change in any one of these factors influences the endurance and cushioning values. It has been erroneously believed that two springs having the same stiffness rate statically, would show the same amplitude of movement on the car. This we have disproved. We now know that, in two springs of the same stiffness rate under a static test, the thicker the spring is, the smaller will be its amplitude of movement in consequence of a given impact.

Interleaf Friction

In leaf springs this factor is important as it directly affects the cushion as well as the endurance values. Some friction is desirable but too much is undesirable. It is, of course, advisable to keep the friction reasonably constant.

Fig. 15 shows a standard road scale called the lodometer, which is used to determine the ground weight on each wheel of the vehicle. The wheel is driven up on it as it lies on the ground.

Fig. 16 is an engineer's report covering the suspension. Here are shown the nine main factors recorded in the Ride-O-Graph test. They indicate the effectiveness of a given combination of springs, tires and shock absorbers. Factor No. 1 pertains to springs only, the other eight pertain to the combined effect of springs, tires and shock absorbers. In arriving at a decision as to the status of a given factor from observing and analyzing the cars, a standard practice has

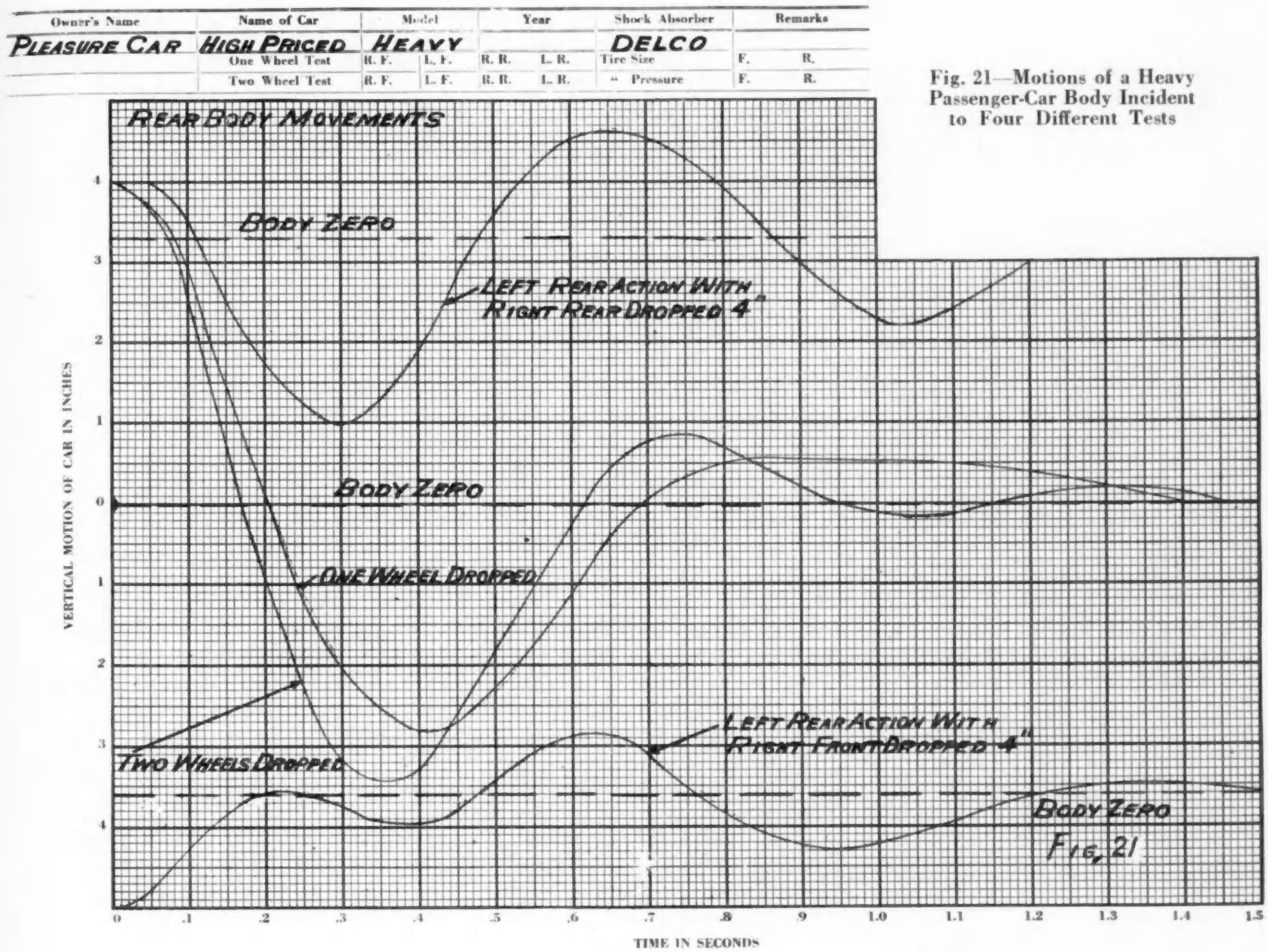


Fig. 21—Motions of a Heavy Passenger-Car Body Incident to Four Different Tests

been established. A check mark is made at one of the three letters L, M and H appearing under each factor. L, means low; M, medium; and H, means high. The last letter under each factor indicates the most undesirable condition. Therefore, such a condition is susceptible to the greatest improvement.

Fig. 17 shows an accessory devised so as to hold the axle to the chassis during the drop test, as the axle tends to drop faster than the body because of the energy stored up in the spring when the body weight is upon it. The hook at the top is rested on the chassis channel and the lower arm is placed under the axle. Then the nut at the bottom is screwed up until the clamp is firm under the axle. The instant the wheel hits the ground, this clamp releases and the small coil spring throws it out and it disengages, thus giving the axle freedom of movement. This test simulates the tire striking a raised irregularity in the road with the body weight upon it. After analyzing many of these tests and comparing them with the free drop tests, the information secured as a result of using this device was not of sufficient additional value to warrant continuance of its use. Realizing that the tire and spring motion represented a constant, it became obvious that when we once observed these motions incident to the free drop tests, we were then able to make definite comparisons with different combinations of springs and tires and note the relative differences which existed. However, when the shock absorber enters into the combination, it introduces a variable condition.

Fig. 18 shows three curves which we have set up as standards, covering the body motions on medium weight passenger cars at their front ends. There are three sets of these standard curves. Another set applies to light weight cars and the other to heavy weight cars. The three curves on each set portray three conditions; poor, good and best. These standardized sheets containing these standard curves are made of transparent paper and are laid over the actual Ride-O-Graph curve so that quick comparisons can be made. The "poor" curve does not represent the poorest we have found, but the average of the poor ones. The "best" curve represents the average of the best group we have thus far tested. All standard curves are on the leaf sprung cars. Coil sprung curves have not as yet been established.

Fig. 19 shows the three standard curves we have established for the rears of medium weight cars. There are three sets of these curves covering rears, the same as on the fronts, one for light cars, one for medium and one for heavy cars. The main differences as between front and rear curves, are the oscillation rates and the accelerations, the rears being much lower in both factors. In rare cases we find passenger cars where the oscillation rates and accelerations are higher in the rear than in the front. In heavy commercial vehicles, the opposite is the case because the spring stiffness and the tire deflection rates in most of them are higher in the rear than in the front.

The seven curves in Fig. 20 represent typical complaints and causes and greatly expedite the analysis of the Ride-O-Graph curves and facilitate a decision as to the remedy which will rectify the complaint. The explanation opposite each

curve tells tersely the cause of and effect produced by each of these deficient body motions.

The four curves in Fig. 21 show the motions of a heavy passenger car body incident to four different tests. One shows a 4-in. test with both rear wheels dropped; another with one rear wheel dropped; another with the right rear wheel dropped, showing the body motion at the opposite side, and another with the right front dropped, showing the body motion at the left rear.

A new air shock absorber just coming on the market has been installed in connection with several bus clinics in combination with special customs springs. This air device is unlike any other shock absorber on the market. It requires no change in the standard chassis incident to its installation, nor does it require that air pressure be maintained, as it takes its air from atmosphere and builds up its own pressure to meet the desired shock requirement. It is double acting and adjustable on compression and rebound.

Engineers in designing chassis springs on commercial vehicles must first figure on a spring of a sufficient stiffness rate to carry the maximum static load. They then usually add 50 to 100 per cent to provide a factor of safety and take care of the dynamic road shock requirements. With this air device, we were able in some cases to cut the stiffness rate of the standard factory equipment springs almost in half, particularly in the rear. Many road tests are now being made with vehicles containing this combination and it is believed that gratifying reductions in maintenance costs will be reported before very long.

Fig. 22 shows a test on a 30 passenger bus. The high accelerations of the body are caused by the axle. During the body motion downward the axle accelerated upward at a rate of 945 ft. per sec. per sec. which caused the body coming downward to slow up at a deceleration rate of 560. Then the body accelerated upward at a rate of 367 ft. per sec. per sec. When the axle reached its maximum rebound and started downward, it jerked the body downward at a rate of 1120 ft. per sec. per sec. The body acceleration as it reaches its maximum downward motion, is very high, deceleration being 103 and acceleration being 123. Such high accelerations when they exist in a vehicle are extremely damaging to all of its front end units. The high wheel deflection indicates high tire stresses. The high accelerations indicate high tire wear. The high rebound indicates unsafe steering.

The chart in Fig. 23 shows three curves taken at the front of a popular $1\frac{1}{2}$-ton truck. One curve shows the standard factory equipment spring; another a custom spring with no control; another with a rebound control attached to the custom spring. The improvements over standard, are very apparent. Custom spring with the rebound control, damped out the body oscillations much sooner. The accelerations on the first rebound were reduced over 50 per cent, on the second deflection 65 per cent, and on the second rebound 68 per cent. Amplitudes were cut down materially. The axle amplitudes and accelerations were materially improved by the custom spring with its control attachment.

Summation

Our objective during all these activities has been to develop a reliable means which would disclose hidden facts pertaining to the suspension; facts which for years the industry has admitted were important but which it had been unable to evaluate.

In so far as tests pertaining to the important values in springs, tires and shock absorbers are concerned, we feel that our objective has been accomplished in the main. It remains

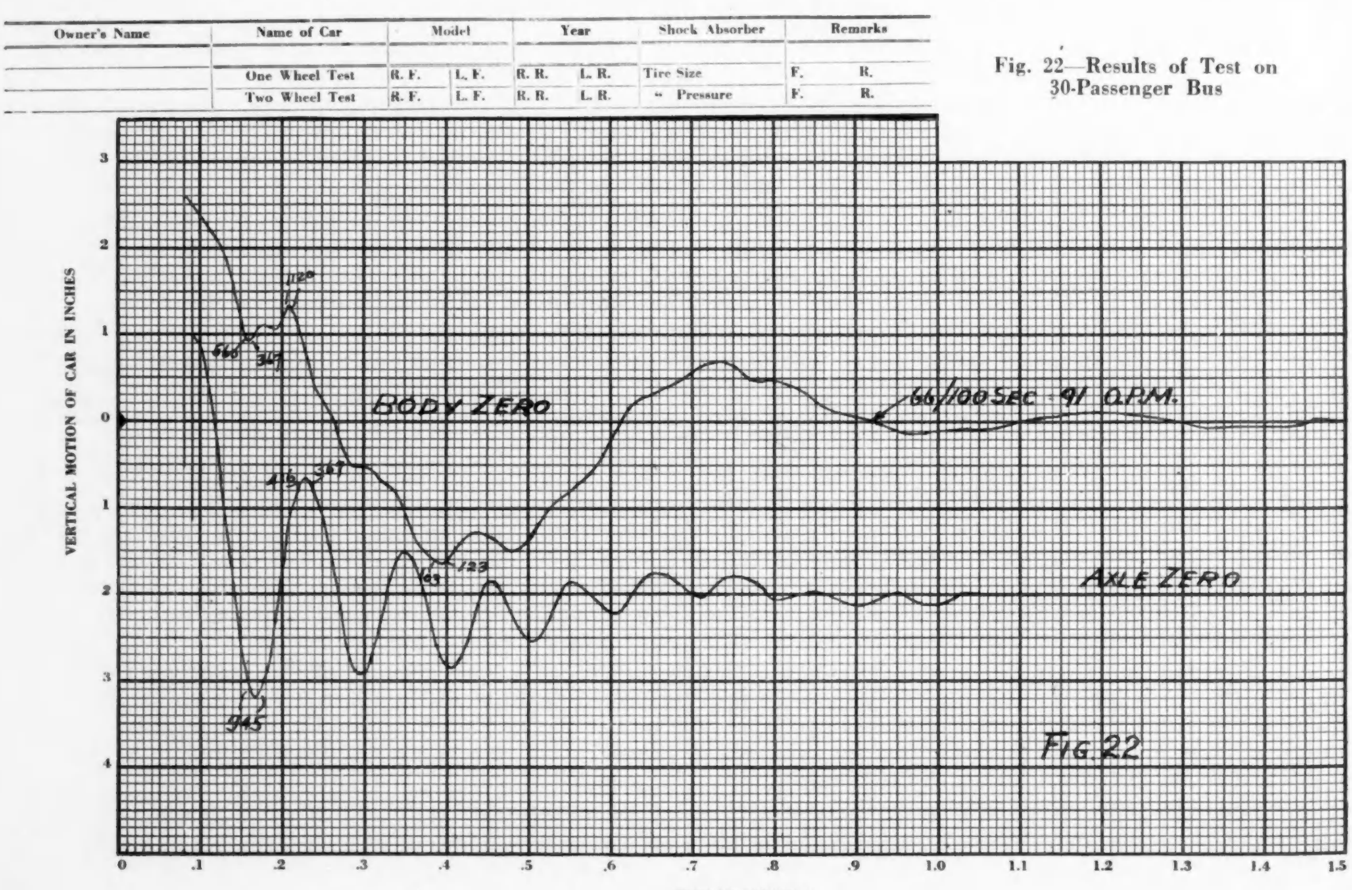


Fig. 22—Results of Test on 30-Passenger Bus

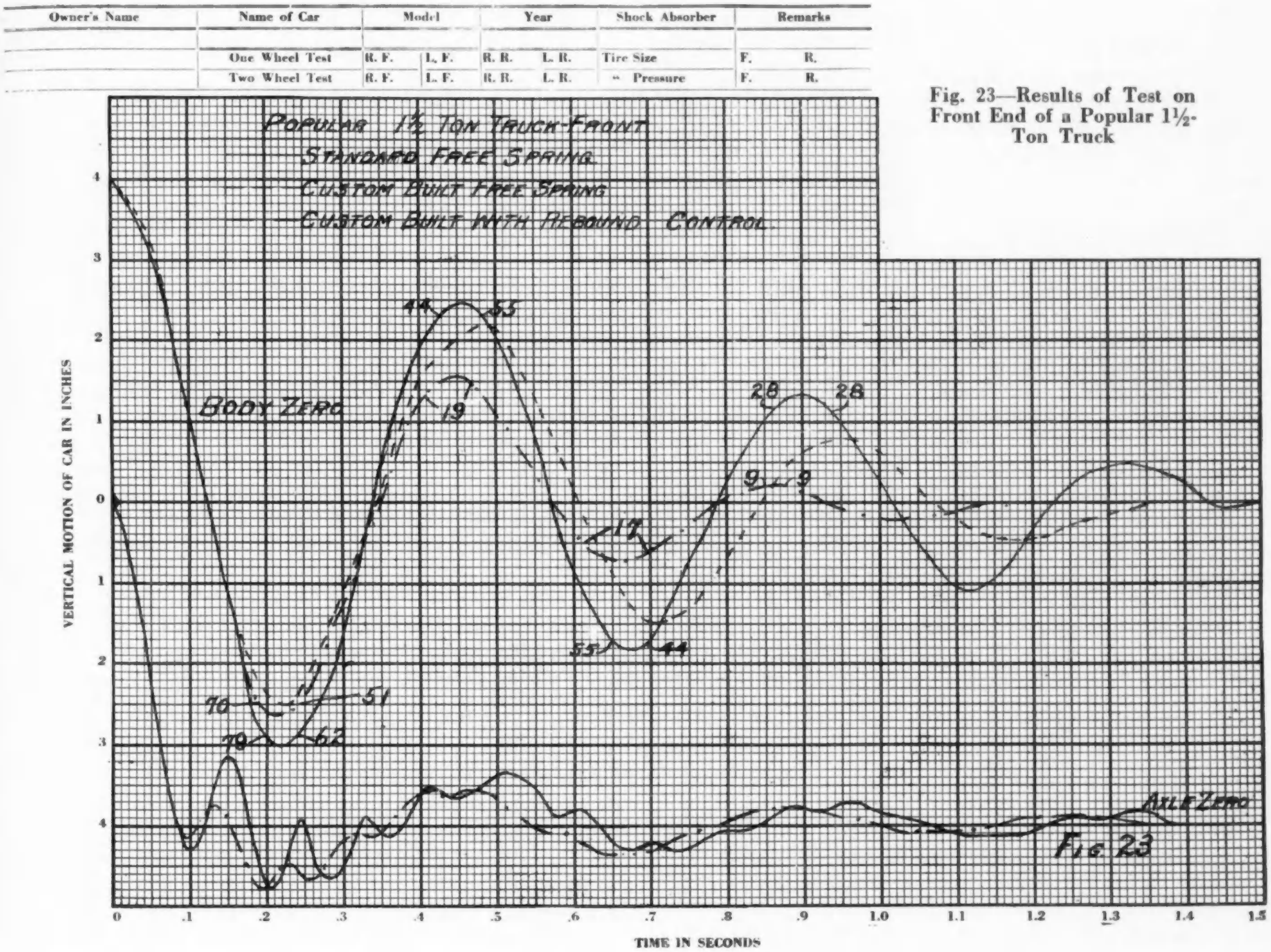


Fig. 23—Results of Test on Front End of a Popular 1½-Ton Truck

to be seen whether or not the seller of these elements, desires to give his buyer definite figures which will indicate to the buyer just what each element, or the suspension as a whole, will produce in cushioning and safety effectiveness. On the other hand, will the buyer be interested in these figures as he is now in horsepower, pick-up and braking efficiency? In other words, will the buyer be inclined to scrutinize vertical acceleration figures the same as he now scrutinizes horizontal acceleration figures so as to know definitely what the vehicle produces in comfort qualities, in the case of a passenger car or what its anticipated costs of operation will be, in the case of a commercial vehicle? Time only will tell.

In conclusion, I want to give credit to Walter F. Whiteman, my worthy collaborator in this work. Mr. Whiteman has handled all the mathematics and engineering details and much of the credit for the results accomplished, is due to his able and untiring efforts. His abundant experience in this field has also been most helpful.

Appendix by W. F. Whiteman

Referring to Fig. 5, we found in our endeavor to make the process easier for the layman to understand, that it became more difficult for the engineer to understand. In the commercial field, we decided it was advisable to make the acceleration curve a discontinuous one. Therefore, we plotted the scales of the acceleration above the abscissa and the deceleration below it, thereby eliminating minus signs. Velocity is similarly treated. It also is shown with velocities downward, plotted below the abscissa and velocities upward, above the abscissa.

Regarding the method we adopted for quick determination of the accelerations, at the peaks of the curves, where the highest usually occur; we first ascertain what radius of a circle will fit the left side of the peak of the curve under observation. We then locate the tangent point where the radius of the circle and the Ride-O-Graph curve merge. We then determine the average acceleration of the motion during the interval of time the motion consumed in traveling from the point where the two curves merge until the motion reached the point of maximum deflection, which is the point of zero velocity. This interval of time on the Ride-O-Graph curve, is represented by the horizontal distance between the point where the radius and Ride-O-Graph curves merge, and a vertical line through the point of maximum deflection. This interval in different curves varies from 2½/1000 to 2/10 of a second.

The figures obtained by this procedure represent the average minus acceleration during the interval when the motion is approaching its maximum minus acceleration. We then determine the acceleration of the motion as it departs from maximum deflection (zero velocity) as shown on the right side of the peak of the curve, following the same procedure on this side as we do on the left. Almost invariably, maximum acceleration during the cycle, occurs at the peak of the curve. One of the few exceptions is shown in Fig. 22.

The Ride-O-Graph curves rarely are pure sine curves. Most of them are greatly distorted, particularly at their peaks. In

a sine curve, the deceleration and acceleration rates at their peaks, are theoretically uniform. However, in the Ride-O-Graph curves where the actual motion of the body and axle are revealed, the deceleration rate in some cases is much higher than the acceleration rate at the peak of the curve. In other cases, the acceleration rate is much higher than the deceleration rate. Frequently, one is several times greater than the other. Therefore, it is seen that the theoretical maximum acceleration rate of a given sine curve may be greatly higher or lower than the average acceleration figure as we determine it, from the actual curve. In numerous comparisons made, using the figure obtained by the mathematical calculation procedure, where it is assumed the curve is a pure sine curve and comparing this figure with the average acceleration figure as we determine it, these figures sometimes vary as much as 5 to 1. From experience gained in consequence of analyzing thousands of these curves which reveal the exact motion of the vehicle, there is no question but what the method we have adopted produces the most reliable figure. We have proved that these figures thus attained are definite and reliable, also commercially practical on thousands of vehicles that have been tested and analyzed.

The formula used in determining the average acceleration at the peaks of the Ride-O-Graph curves, is:

$$a = \frac{7.42}{r \cos \theta}$$

This formula is based on the basic acceleration formula:

$$a = \frac{V_2 - V_1}{t}$$

Tables have been set up and transparent templates made for various radii, from 1/64 to 2.0 in., inclusive. On these templates, the angles covering each radius from 5 deg. to 85 deg., inclusive, are indicated. By using the particular template which fits the radius on the Ride-O-Graph curve at its peak, and then noting the angle of the mutual tangent on the template, the acceleration is read. This method is equivalent to graphically differentiating the space-time curve, which takes the increment as the distance between a point on the tangent where the Ride-O-Graph curve merges with the circle on the template, and the point of maximum deflection or rebound on the Ride-O-Graph curve.

For the mathematical determination of the instantaneous acceleration, the equation is:

$$a = \frac{d^2s}{dt^2} = be^{-ct}\left[(c^2 + f^2)\sin ft - 2cf\cos ft\right]$$

which is the second derivative of the oscillation curve of decreasing amplitude:

$$s = be^{-ct}\sin ft$$

The sine curve for harmonic motion which is most used, is:

$$s = b \sin ct$$

$$\text{and} \quad a = \frac{d^2s}{dt^2} = -bc^2 \sin ct$$

Mr. Widney, in speaking of shock absorbers, stated that the spring was the main shock absorbing or cushioning element between the sprung and unsprung mass. Cushioning may be defined as that property of a material or mechanism which enables the material or mechanism to dissipate energy that has been imparted to it. Its cushioning effectiveness is measured by its ability to minimize the instantaneous force. In other words, the amount of cushion which the cushioning element affords, depends upon its ability to distribute the applied force evenly through the space in which it acts.

Discussion of D'Alleva-Lovell Paper

(Continued from page 98)

the experimental data were obtained. The percentage of the products found is plotted against measured instead of computed air-fuel ratio. This does not seem justifiable when all the assumptions were based on exhaust gas analysis. Inspection of the authors' Fig. 3 shows that this change should not have been made because the computed air-fuel ratio is not equal to the measured air-fuel ratio. The data obtained by the N.A.C.A. (Technical Report 476) show the same result. The linear divergence between computed and measured air-fuel ratio as the mixture becomes richer seems to be what one would expect because as the mixture becomes richer more and more carbon or soot is formed during combustion and can be seen in the exhaust. Chemical analysis of the products of combustion does not take this carbon into account.

The authors' experimental and theoretical data have been replotted in the chart accompanying this discussion on the basis of air-fuel ratio determined from the exhaust. It is apparent from this figure and the authors' Fig. 3 as well as the data reported in N.A.C.A. Technical Report 476 that the air-fuel ratio obtained from a knowledge of the products of combustion is not the air-fuel ratio entering the engine. It should be borne in mind that when using the replotted figure together with a partial analysis of the exhaust gas for determining the volumetric efficiency or air flow, it is necessary to correct the exhaust air-fuel ratio by means of the authors' Fig. 3 to intake air-fuel ratio.

As the authors experimented with only one fuel it is not known what the correlation would be for other fuels. However, Gerrish and Tessmann (N.A.C.A. Report 476) found a different amount of methane present when using fuels of considerably different hydrogen-carbon ratio.

The authors' claim that the presence of carbon monoxide and hydrogen in the exhaust gas at air-fuel ratios greater than 14.7 is due principally to imperfect fuel distribution between the various cylinders. It seems desirable to call attention to the information published in N.A.C.A. Technical Report 476 wherein it is shown that these gases are present in this region for two different single-cylinder engines using gasoline having a hydrogen-carbon ratio of 0.183 by weight. It is believed that the presence of these gases is due more to the lack of a perfectly homogeneous charge in the cylinder than to an unevenness in distribution between the cylinders.

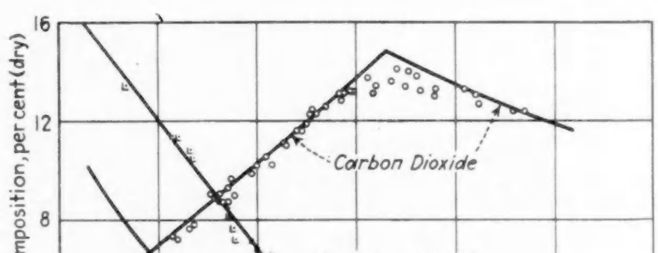


Fig. A—(Gerrish Discussion) Experimental and Theoretical Exhaust Gas Relations